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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

REPORT No. 577

PRECHAMBER COMPRESSION-IGNITION ENGINE PERFORMANCE

By CHARLES S. MOORE and JOHN H. COLLINS, JR.

1937

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AERONAUTIC SYMBOLS

1. FUNDAMENTAL AND DERIVED UNITS

	Symbol	Metric		English	
		Unit	Abbreviation	Unit	Abbreviation
Length	l	meter	m	foot (or mile)	ft. (or mi.)
Time	t	second	s	second (or hour)	sec. (or hr.)
Force	F	weight of 1 kilogram	kg	weight of 1 pound	lb.
Power	P	horsepower (metric)		horsepower	hp.
Speed	V	kilometers per hour	k.p.h.	miles per hour	m.p.h.
		meters per second	m.p.s.	feet per second	f.p.s.

2. GENERAL SYMBOLS

W, Weight $= mg$

g, Standard acceleration of gravity $= 9.80665$ m/s^2 or 32.1740 ft./sec.^{2}

m, Mass $= \frac{W}{g}$

I, Moment of inertia $= mk^2$. (Indicate axis of radius of gyration k by proper subscript.)

μ, Coefficient of viscosity

ν, Kinematic viscosity

ρ, Density (mass per unit volume)

Standard density of dry air, 0.12497 kg-m^{-4}-s^2 at 15° C. and 760 mm; or 0.002378 lb.-ft.$^{-4}$ sec.^{2}

Specific weight of "standard" air, 1.2255 kg/m^3 or 0.07651 lb./cu.ft.

3. AERODYNAMIC SYMBOLS

S, Area

S_w, Area of wing

G, Gap

b, Span

c, Chord

$\frac{b^2}{S}$, Aspect ratio

V, True air speed

q, Dynamic pressure $= \frac{1}{2}\rho V^2$

L, Lift, absolute coefficient $C_L = \frac{L}{qS}$

D, Drag, absolute coefficient $C_D = \frac{D}{qS}$

D_o, Profile drag, absolute coefficient $C_{D_o} = \frac{D_o}{qS}$

D_i, Induced drag, absolute coefficient $C_{D_i} = \frac{D_i}{qS}$

D_p, Parasite drag, absolute coefficient $C_{D_p} = \frac{D_p}{qS}$

C, Cross-wind force, absolute coefficient $C_C = \frac{C}{qS}$

R, Resultant force

i_w, Angle of setting of wings (relative to thrust line)

i_t, Angle of stabilizer setting (relative to thrust line)

Q, Resultant moment

Ω, Resultant angular velocity

$\frac{\rho Vl}{\mu}$, Reynolds Number, where l is a linear dimension (e.g., for a model airfoil 3 in. chord, 100 m.p.h. normal pressure at 15° C., the corresponding number is 234,000; or for a model of 10 cm chord, 40 m.p.s. the corresponding number is 274,000)

C_p, Center-of-pressure coefficient (ratio of distance of $c.p.$ from leading edge to chord length)

α, Angle of attack

ϵ, Angle of downwash

α_o, Angle of attack, infinite aspect ratio

α_i, Angle of attack, induced

α_a, Angle of attack, absolute (measured from zero-lift position)

γ, Flight-path angle

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By Charles S. Moore and John H. Collins, Jr.

SUMMARY

Single-cylinder compression-ignition engine tests were made to investigate the performance characteristics of the prechamber type of cylinder head. Certain fundamental variables influencing engine performance—clearance distribution, size, shape, and direction of the passage connecting the cylinder and prechamber, shape of prechamber, cylinder clearance, compression ratio, and boosting—were independently tested. Results of motoring and of power tests, including several typical indicator cards, are presented.

Results of the investigation indicate that for maximum performance of this 5- by 7-inch engine at speeds up to 1,500 r. p. m., the compression ratio should be between 15.5 and 17.5 and the prechamber should be as large as possible, disk-shaped, and connected to the cylinder by a single passage. A strong rotational air flow should be created in the prechamber by introducing the passage tangentially. Flaring should be employed on the cylinder end of the passage to spread the issuing gases over the flat piston crown. At 1,500 r. p. m., the injection system should deliver in approximately 20 crankshaft degrees the full-load fuel in the shape of a narrow conical spray with high penetration. This spray should be directed across the disk chamber toward the mouth of the connecting passage. Boosting the inlet-air pressure effectively raises the power output. As the prechamber is inaccessible for scavenging and the lack of clearance under the valves prohibits the use of proper valve timing, the prechamber type of cylinder head is judged to be incapable of developing the high specific output required of aircraft engines.

INTRODUCTION

The general problem in the development of aircraft compression-ignition engines is to obtain complete and properly timed combustion in the engine cylinder at high crankshaft speeds. A prime requirement for complete combustion is that the fuel charge be intimately mixed with the air. Furthermore, combustion must be so controlled that it is completed early in the power stroke without combustion shock. In order to accomplish these requirements, numerous chamber designs and fuel-spray arrangements have been tried by different designers with varying degrees of success. Each has its own relative merits and its own field of usefulness.

The prechamber, which may be classed as an auxiliary-chamber type, has been extensively used. Its popularity is no doubt due to the simplicity of the fuel-spray arrangement which may be used and to the variety of means which may be employed to control the mixing and combustion of the fuel and air. The auxiliary chamber may function as an air reservoir to meter the air to the cylinder, or it may serve as a mixing chamber in which the fuel charge is prepared for combustion before it passes into the cylinder. When combustion starts and is partly completed in the auxiliary chamber, this type becomes the usual precombustion chamber.

For designs in which the auxiliary chamber acts as a prechamber, or mixing chamber, the connecting passage and chamber have two functions to perform: First, the forced air flow is controlled by the size, shape, and direction of the connecting passage, these factors being selected to give the best mixing in the chamber with the least loss by resistance to the flow; and second, the mixing of the fuel and air is controlled by the size, shape, and position of the prechamber, these factors being designed to conserve and utilize the forced air flow as a residual flow. After combustion starts, the passage further functions to meter and direct the partly burned, overrich mixture into the cylinder in such a way that all the cylinder air is reached by the unburned fuel and at such a rate as to control the pressures developed in the cylinder.

As a part of a general research on aircraft-type compression-ignition engines, the Committee has been investigating the performance to be obtained with the prechamber type of cylinder head. Most of the work herein reported has been published as the several investigations were completed; the purpose of the subject report is to include the final and unreported work of the investigation and to combine all the more important results into a single publication.

APPARATUS AND TEST PROCEDURE

TEST ENGINE

The single-cylinder-engine test unit shown in figure 1 was used in this investigation. This figure shows the assembly of equipment at the time the investigation was completed; the original set-up, however, differed only in minor details. The compression-ignition 4-

stroke-cycle engine had a 5-inch bore and 7-inch stroke. Originally a single-cylinder Liberty test engine was used (reference 1), in which the cast-iron head was bolted to a special steel cylinder; the cylinder in turn was bolted to the engine crankcase. Later a more flexible unit was required, and an N. A. C. A. universal engine crankcase (reference 2) and cylinder were substituted for the Liberty crankcase and cylinder. Standard Liberty engine parts were used wherever possible. Fuel, oil, and water temperatures were maintained at 80°, 140°, and 170° F., respectively, during all tests. A 50–75 horsepower electric cradle-type dynamometer measured the torque and absorbed the engine power.

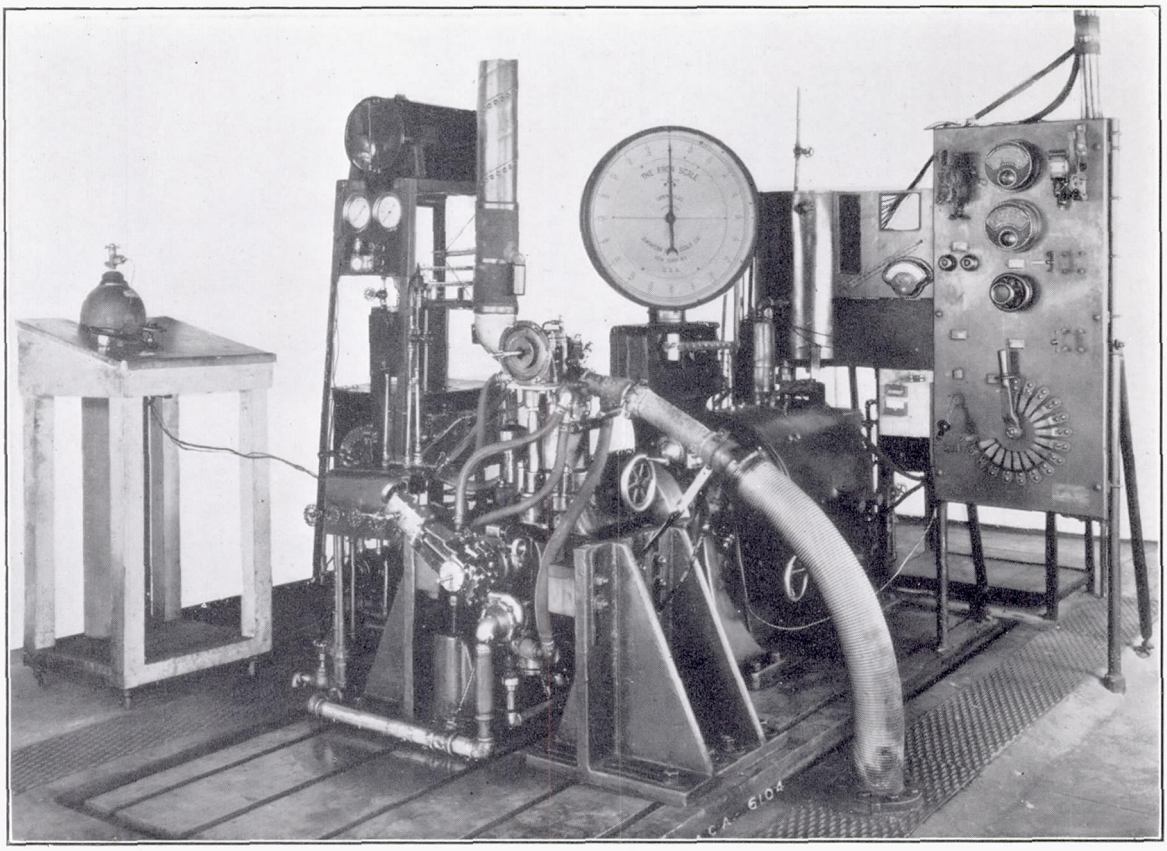


FIGURE 1.—Single-cylinder engine and test equipment.

CYLINDER HEADS

The several cylinder heads used in this investigation will be described in the order of their use. In the first design a pear-shaped auxiliary chamber was cast integrally with the head with a 1-inch-diameter passage connecting the chamber to the cylinder. With a standard high-compression Liberty piston the compression ratio was 9.9. The first combustion-chamber design was altered in order to increase the compression ratio and to create a higher degree of turbulence within the pear-shaped chamber on the compression stroke and within the cylinder on the expansion stroke (fig. 2). The turbulence was generated by locating the $\frac{9}{16}$-inch-diameter passage to produce tangential flow in both the bulb and the cylinder. When the piston was at top center, the ratio of the volume of air in the pear-shaped chamber to the volume of air in the cylinder was approximately 1. Preliminary tests at a compression ratio of 13.5 were made to determine the effect of progressively altering the passage shape.

N. A. C. A. cylinder-head design 7 was made (see fig. 3) to permit a wide range of changes in the connecting passage and auxiliary chamber without disturbing other parts of the head. By the construction and assembly of different chamber parts and adjustments of the universal test engine, this cylinder head was readily adapted to the investigation of a variety of combustion-chamber forms and variables.

AUXILIARY TEST EQUIPMENT

Except for those tests in which the fuel-injection system was the variable, the same injection system was used in all the tests. A speed-reduction and timing mechanism, which operated the pump at camshaft speed, allowed the injection advance angle to be varied while the engine was running by changing the angular relation of the fuel cam with respect to the crankshaft. Varying the duration of the closure of a bypass valve in the constant-stroke pump controlled the quantity of fuel delivered to an automatic fuel-injection valve. A single 0.050-inch-diameter orifice with a length-diameter ratio of 2.5 was used in connection with a plain stem.

The Diesel fuel used in most of the tests had a specific gravity of 0.847 and a viscosity of 41 seconds Saybolt Universal at 80° F. Fuel input was measured by timing electrically the consumption of ½ pound of fuel oil while a synchronized revolution counter recorded the number of engine revolutions. Air consumption was measured by recording the time required for 80 cubic feet to be displaced from a 100-cubic-foot gasometer. Explosion pressures were indicated by the N. A. C. A. balanced-diaphragm valve. Indicator cards were obtained with a Farnboro electric indicator. A stroborama was used to determine the injection periods and injection advance angles. From the indicator cards, the ignition lags and rates of pressure rise were determined. The ignition lag is considered as the time in seconds from the start of injection of the fuel to the start of pressure rise on the card.

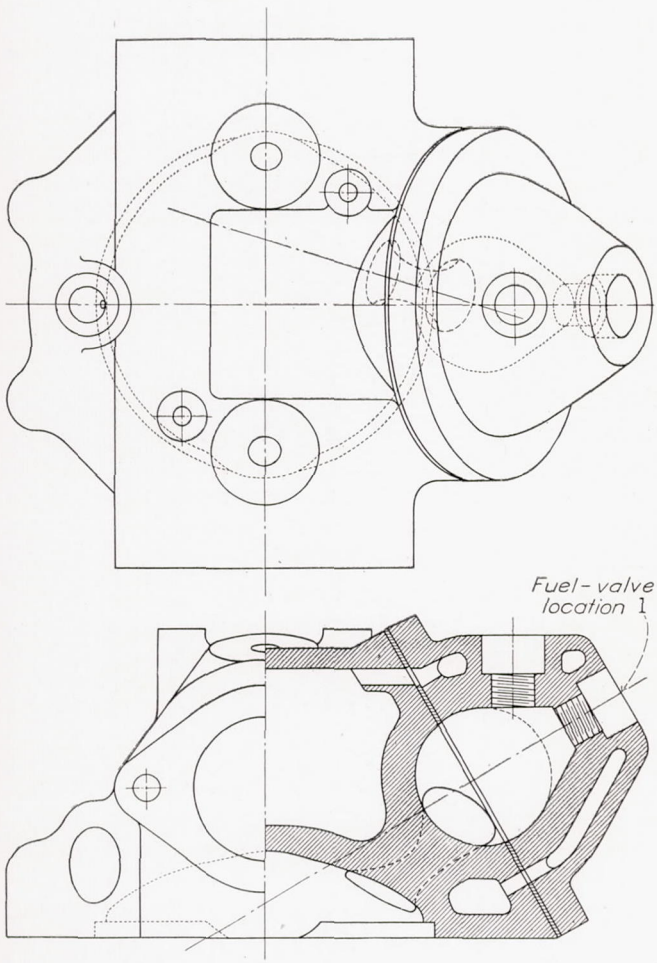
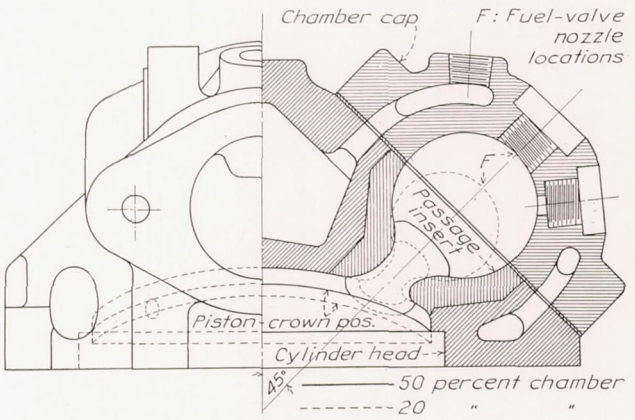


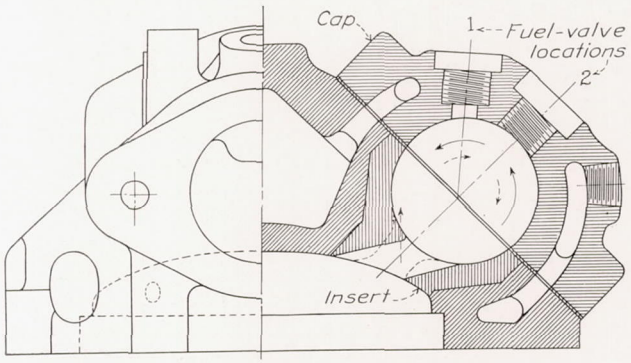
FIGURE 2.—Cylinder-head design showing pear-shaped prechamber.

TEST PROCEDURE

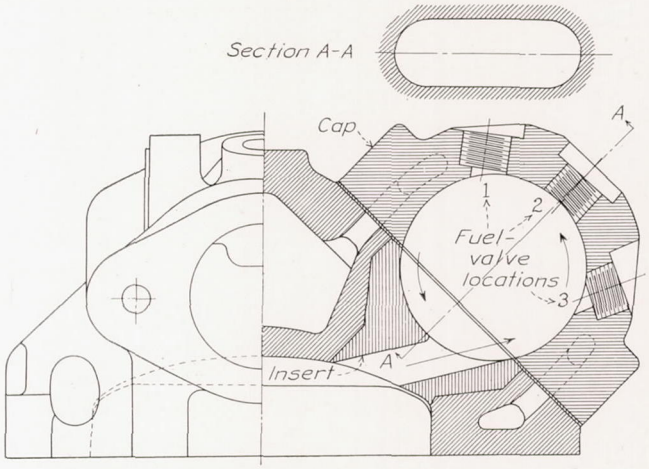
After the preliminary investigation was completed, a more systematic study was undertaken. The most important variables indicated by an analysis of the problem were studied in the following order: clearance distribution between cylinder and chamber, connecting-passage diameter, prechamber shape, cylinder-clearance shape, compression ratio, and boost pressure. Throughout the investigation, only one variable at a time was changed; all other conditions were held constant insofar as was conveniently possible. Although in some cases a dependent factor changed when an attempt was made to change a single variable, in no case was this condition permitted if it was of major importance or if it was economically possible to vary only the single variable. All pieces of apparatus were calibrated at intervals during the tests and corrections were applied to the results. During the testing, the barometric pressure varied from 29.49 to 30.40 inches of mercury; no attempt



(a) Spherical prechamber with radial passage.



(b) Spherical prechamber with tangential and radial (dotted) passage.



(c) Disk-shaped prechamber.

FIGURE 3.—Cylinder-head designs showing different prechambers.

has been made, however, to correct the data to a standard pressure, temperature, or humidity because there is no generally recognized method of correcting compression-ignition data to standard conditions.

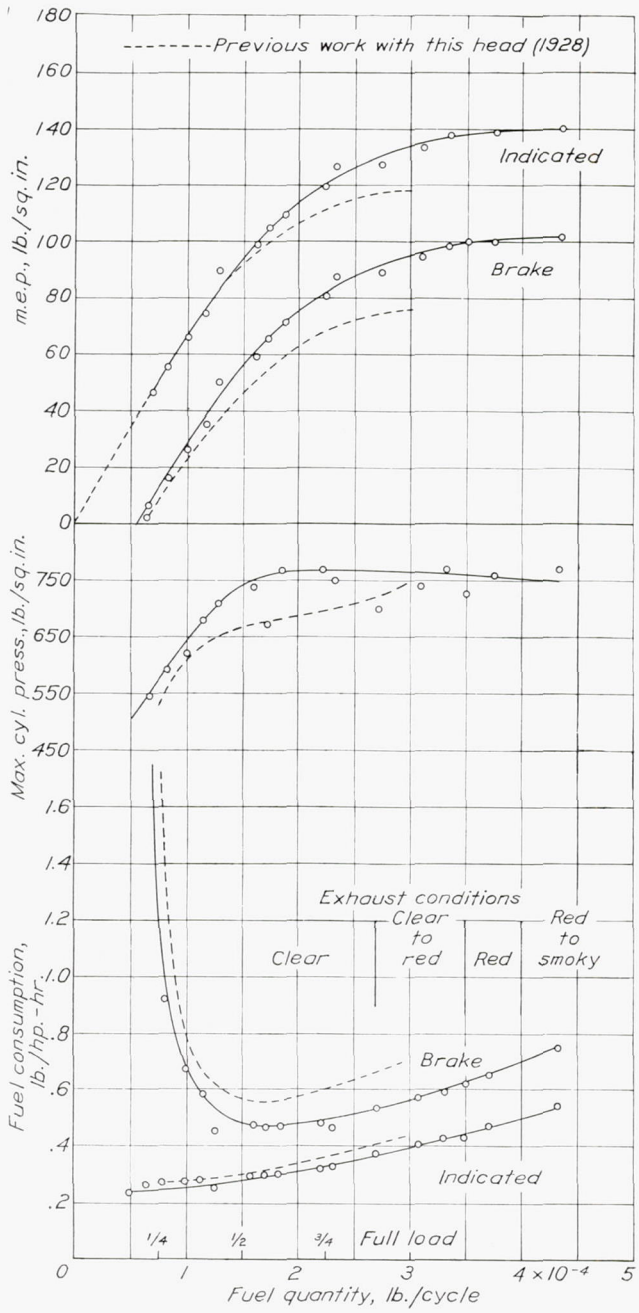


FIGURE 4.—Effect of fuel quantity on engine performance. Engine speed, 1,500 r. p. m.; cylinder head as shown in figure 2; compression ratio, 13.5; N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 2.5.

TESTS AND RESULTS

PRELIMINARY INVESTIGATION

The curves showing the results of the preliminary investigation of the prechamber type of cylinder head are presented in reference 3. Owing to the low compression ratio of 9.9 the engine was difficult to start. Acceptable operation, however, was obtained at 1,800 r. p. m. which, at the time these data were first published, was considered an exceptionally high speed for compression-ignition engines.

According to the results obtained with the first cylinder head, the N. A. C. A. combustion chamber 3 (fig. 2) was designed to cause a residual air flow for mixing the fuel and air (reference 3). This head both improved the performance and reduced the maximum cylinder pressures. A series of tests was made with this cylinder head, in which the location of the fuel nozzle with respect to the walls of the auxiliary chamber was varied. With the fuel valve in location 1, the fuel nozzle was extended into the chamber by increments from the flush position to a point 1½ inches from the wall. Different orifices and spray types were tried, but no improvement in performance was obtained over that with the simple spray in the flush position.

With the cylinder head shown in figure 2, the greatest improvement in performance (fig. 4) was obtained by shortening the injection period from approximately 64 to 21 crankshaft degrees, although a small improvement may have been due to flaring the cylinder end of the connecting passage.

CLEARANCE DISTRIBUTION

As maximum performance was not the first consideration, the shape of the combustion chamber was not selected for best performance but to permit the study of chamber size without introducing a secondary variable. For this reason, the chamber was made spherical to permit varying the allocation of the clearance between cylinder and chamber with a minimum change in the shape of the combustion space. The spherical prechambers plus one-half the connecting passage contained 20, 35, 50, and 70 percent of the total clearance at a compression ratio of 13.5. For convenience of reference, these clearance distributions will be called the 20-, 35-, 50-, and 70-percent chambers. Clearance in the cylinder was formed between the domed cylinder head and the domed piston crown.

The connecting passages were circular in cross section, of constant length-diameter ratio, and were flared at both ends. Each of the four passages was designed to have a cross-sectional area proportional to the prechamber volume. Thus, at the same engine speed for each of the four clearance distributions, the calculated air velocities through the passages were the same. Passage diameters obtained by this method were $^{23}/_{64}$, $^{15}/_{32}$, $^{9}/_{16}$, and $^{43}/_{64}$ inch for the 20-, 35-, 50-, and 70-percent chambers, respectively. The axis of the passage included the center of the spherical chamber and intersected the cylinder axis at an angle of 45°.

The injection-advance-angle range from misfiring to allowable knocking was negligibly affected by clearance

distribution. With the fuel valve in the lower hole of the 20-percent chamber the operating range increased from 12° to 27°, but the power decreased and the smoke and flame of the exhaust increased. The injection advance angle of 7° at 1,500 r. p. m. gave a start of pressure rise that varied from T. C. to 3° A. T. C. for all clearance distributions, as determined by inspection of indicator cards.

Table I presents additional data on the engine-operating characteristics.

TABLE I

GENERAL OPERATING CHARACTERISTICS—CLEARANCE DISTRIBUTION

[Engine speed, 1,500 r. p. m.; fuel consumption, 3.0×10^{-4} lb./cycle; 12 percent excess air; cylinder head as shown in fig. 3 (a); compression ratio, 13.5; N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 2.5]

Operating characteristics	20-percent chamber		35-percent chamber		50-percent chamber		70-percent chamber	
Combustion knock	Dull—Regular		Slight—Irregular		Hard—Irregular		Harder—Irregular.	
Injection range, allowable knock to miss	13° B. T. C. to 1° B. T. C		10° B. T. C. to 2° B. T. C		11° B. T. C. to 1° B. T. C		10° B. T. C. to 1° A. T. C.	
Centrifugal spray compared to noncentrifugal spray.	Knock and performance slightly worse for centrifugal spray.		Knock and performance worse for centrifugal spray.		Knock and performance worse for centrifugal spray.		Knock and performance slightly better for centrifugal spray.	
Lower fuel-valve position compared to upper position.	Performance worse for lower position i. a. a. range 27°.		No change		Performance worse for lower position.		Performance worse for lower position.	
Optimum valve-opening pressure, lb./sq. in.	3,500		5,000		5,000		3,500.	
Carbon deposits	Chamber, much	Cylinder, little	Chamber, soot and "cake"	Cylinder, soot	Chamber, soot	Cylinder, soot	Chamber, soft soot	Cylinder, soft soot
Cyclic variation in maximum explosion pressure, lb./sq. in.	300	100	200	100	140	130	90	140

Figure 5 shows that, for the design of prechambers used in these tests, the minimum volume in the chamber for good performance is about 35 percent of the total clearance volume. The inferior performance with the 20-percent chamber cannot be attributed to the deposit of fuel on the walls by the noncentrifugal spray because the centrifugal spray that had insufficient penetration to hit the walls gave slightly worse performance. More power was obtained with the larger chambers because of the greater quantity of air ready for initial combustion. Air in the cylinder, being distributed over the piston crown, cannot be effectively reached by the unburned gasses issuing from the chamber and therefore does not materially assist the combustion process.

The motoring and combustion characteristics are shown in figure 6. The motoring characteristics remain nearly constant as the clearance distribution varies. With the smaller chambers, less air is moved through the passage and the friction mean effective pressure should be less; the decrease in friction mean effective pressure, however, is slight. Maximum indicated compression pressures are slightly higher in the chambers than in the cylinder, probably owing to the method used in measuring the pressures (reference 4).

Figure 6 also shows that clearance distribution does not have an appreciable effect on ignition lag. This result may be expected as the conditions of temperature, pressure, and air speed were held constant during the tests. For all clearance distributions the pressure rises are straight lines and of such high rates that it is impossible to measure them accurately; the numerical values are therefore only approximations. As the chamber proportion increases, the chamber rate tends to decrease and the cylinder rate to increase and then to decrease. The larger chambers containing more air should give a faster rate of pressure rise because the fuel and air mixture would have more nearly the correct proportions for complete combustion. The opposite occurs, however, indicating that the passage size influences the rate of pressure rise, the larger passages of the larger chambers allowing the gases to pass more freely into the cylinder.

Improvement in exhaust conditions that occurs with increase of chamber proportions is caused by the availability of more air for combustion in the auxiliary chamber. Decrease in the rate of improvement with increased allocation of clearance to the chamber of more than 35 percent is due to the combination of spray shape and air flow as used in these combustion-chamber forms. This combination allows a maximum of approximately 35 percent of the fuel to be mixed with air for efficient combustion. The remaining fuel is burned either very late or not at all.

An increase in chamber volume from 20 to 70 percent causes the total heat loss to the cooling water to increase from 21 to 29 percent, owing to the increased quantity of fuel burned in the chamber and also to an increase of approximately 10 percent in the total combustion-chamber surface area (table II). The amount of heat loss from the chamber increases with chamber volume and surface, whereas the amount of heat loss from the head decreases. As the combustion in the chamber increases with increased chamber proportion, the cylinder heat loss decreases.

TABLE II

EFFECT OF CLEARANCE DISTRIBUTION ON HEAT LOSS TO COOLING WATER

[Engine speed, 1,500 r. p. m.; fuel consumption, 3.0 × 10⁻⁴ lb./cycle; 12 percent excess air; cylinder head as shown in fig. 3 (a); compression ratio, 13.5; N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 2.5.]

Chamber volume in percentage of total clearance	Distribution of heat loss to cooling water			Percentage of total heat in fuel
	Cylinder	Head	Chamber cap	
	Percent	*Percent*	*Percent*	
20	50	44	6	21
35	46	34	20	22
50	44	20	36	26
70	43	17	40	29

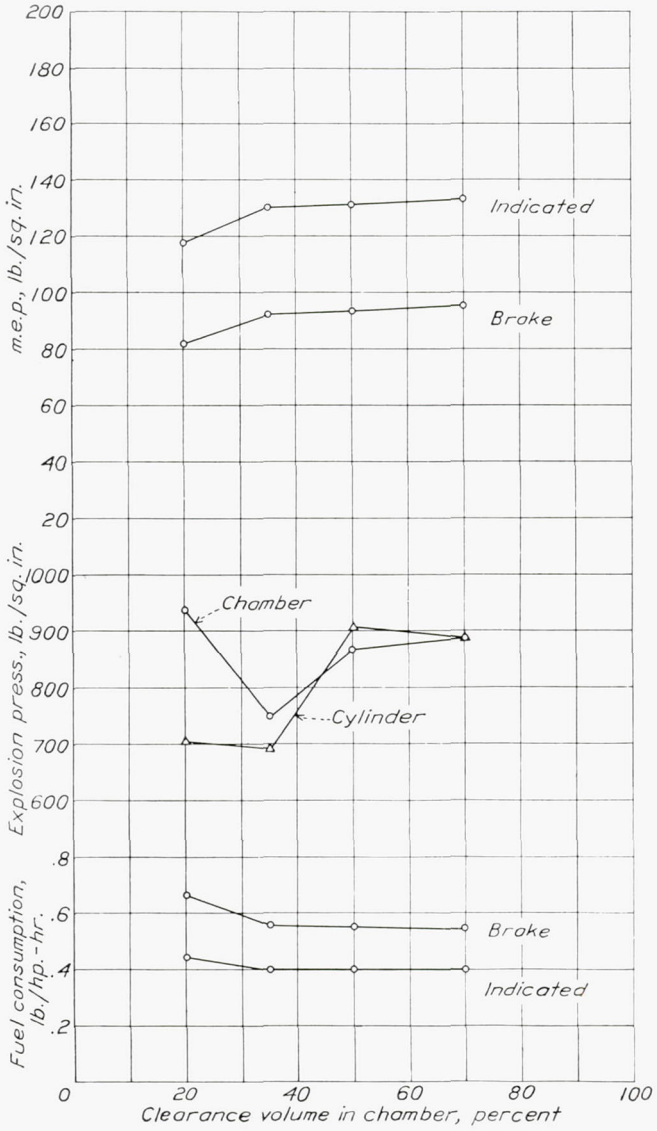


FIGURE 5.—Effect of clearance distribution on engine performance. Engine speed, 1,500 r. p. m.; fuel consumption, 3.0 × 10⁻⁴ lb./cycle; 12 percent excess air; cylinder head as shown in figure 3 (a); compression ratio, 13.5; N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 2.5.

Figure 7 shows the general effect of engine speed and air-flow speed on mean effective pressure and fuel consumption. The trend is nearly the same for all the chambers with the optimum speed, based on maximum i. m. e. p., at 1,200 r. p. m. The larger chambers, because of a more intimate mixture of a larger quantity of fuel and air, developed the most power with the best fuel economy. It is believed that the great difference in i. m. e. p. shown by the curves for the 20-percent chamber was caused by insufficient air in the small

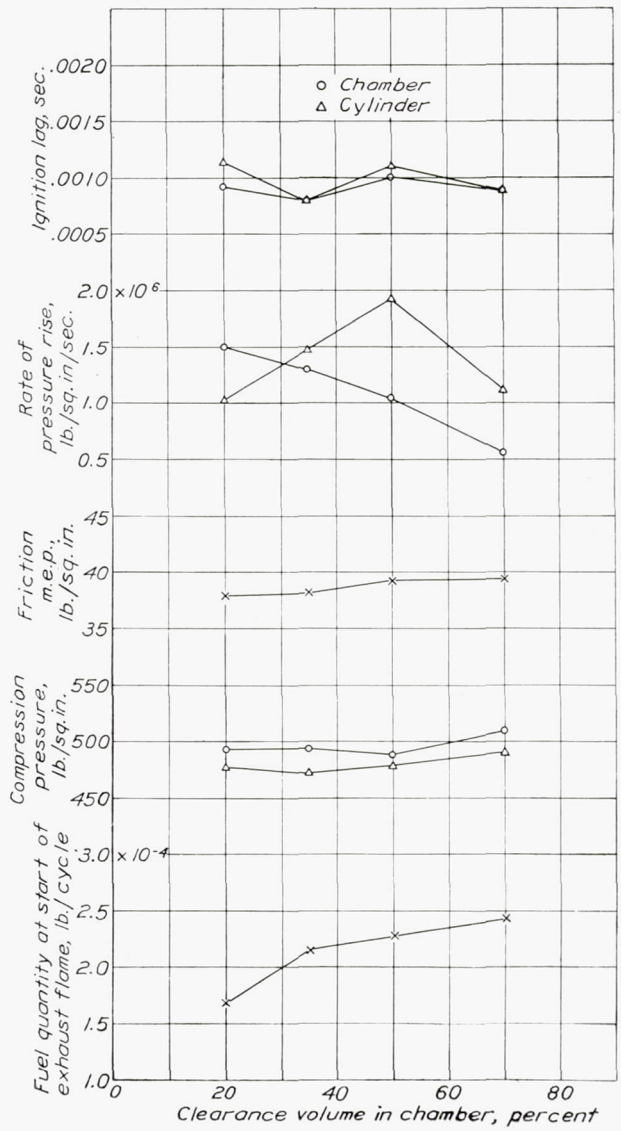


FIGURE 6.—Effect of clearance distribution on motoring and combustion characteristics. Engine speed, 1,500 r. p. m.; cylinder head as shown in figure 3 (a); compression ratio, 13.5; N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 2.5; injection advance angle, 7° B. T. C.

chamber. Explosion pressures of all the chambers increase with speed up to 1,200 r. p. m. because of the better mixing of fuel and air and resultant faster burning. As the engine speed increases above 1,200 r. p. m., most of the curves show a tendency to fall off. The smaller chambers with small passage areas confine the pressure, giving high chamber and low cylinder pressures.

The 50-percent chamber was selected as being representative of all the chambers and the effect of speed on combustion characteristics was investigated. Indi-

cator cards from the other three combustion chambers gave trends similar to those shown in figure 8, which is for the 50-percent chamber. As the engine speed increases, the velocity of air flow in the passage in-

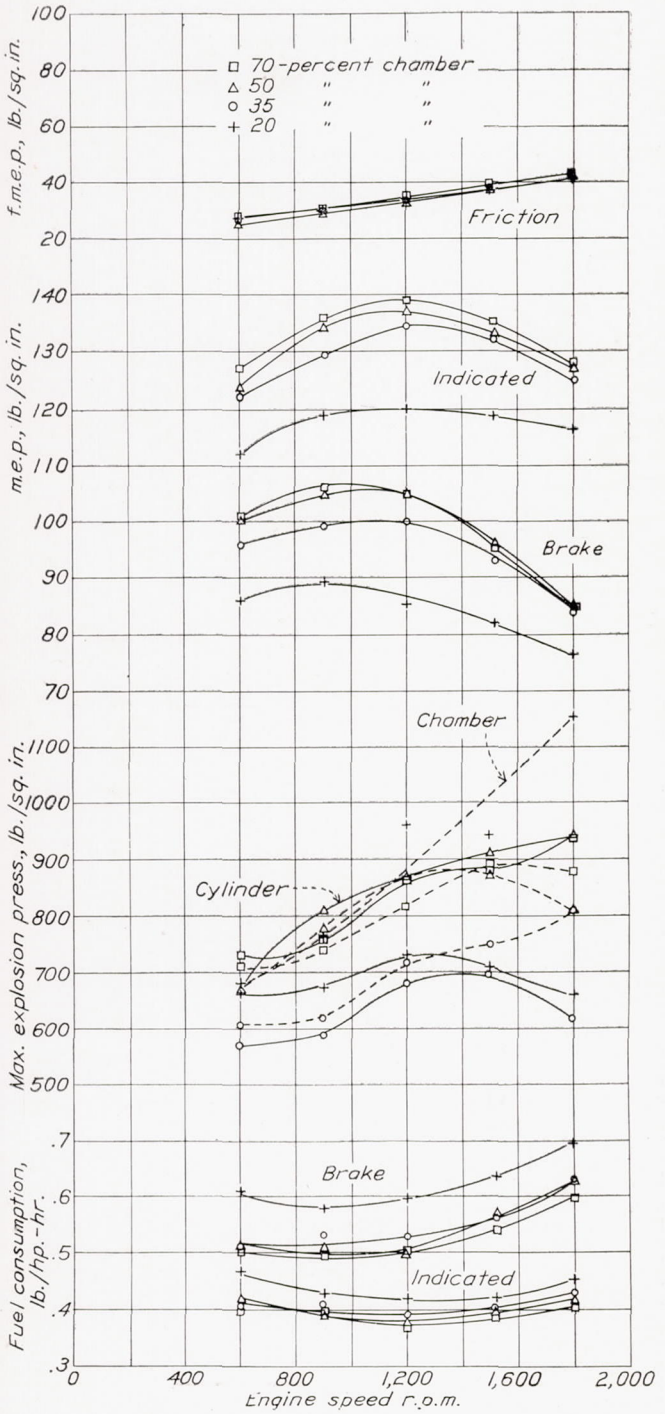


FIGURE 7.—Effect of speed on engine performance. Fuel consumption, 3.0×10^{-4} lb./cycle; 12 percent excess air; cylinder head as shown in figure 3 (a); compression ratio, 13.5; N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 2.5.

creases and the mixing of fuel and air in the chamber is more complete with more rapid combustion and higher rates of pressure rise. Successive engine cycles varied, as the engine sound clearly indicated, so that the points on the Farnboro indicator cards are widely dispersed, especially at the pressure peaks. The rates of pressure rise were obtained by considering the leading points of the card. Apparently, rate of pressure rise and knock do not vary together, inasmuch as the rates of pressure rise were less at the lower speeds and the combustion-knock audibility remained constant.

The starting point of the pressure rise was dependent upon ignition lag and injection advance angle, the latter

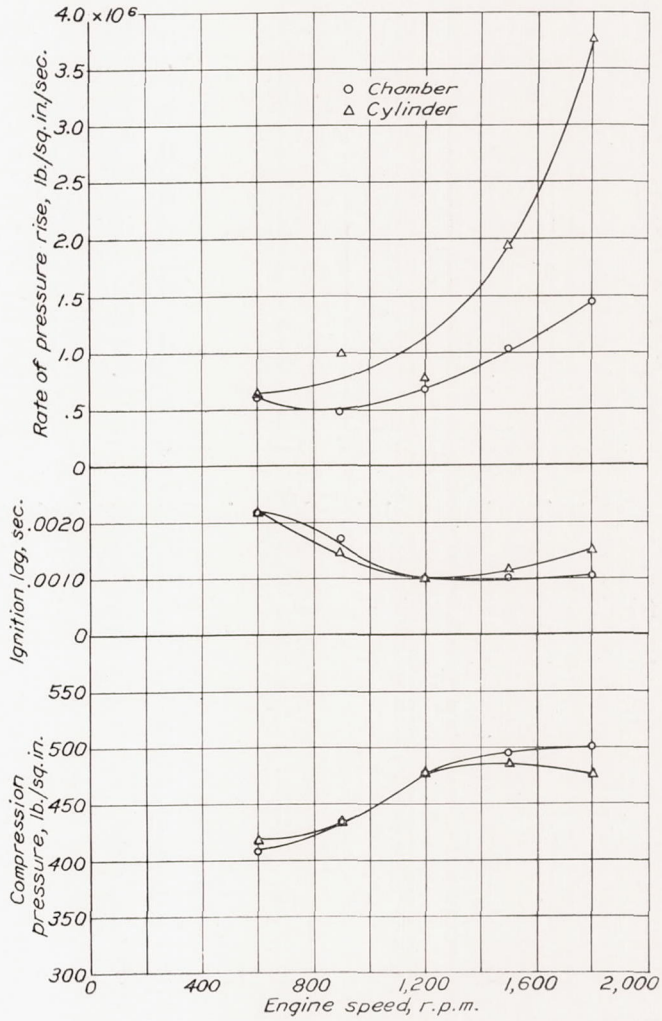


FIGURE 8.—Effect of speed on compression pressure and combustion characteristics. Fuel consumption, 3.0×10^{-4} lb./cycle; 12 percent excess air; cylinder head as shown in figure 3 (a); 50-percent chamber; compression ratio, 13.5; N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 2.5.

being the greatest permitted by allowable knock intensity. The start of pressure rise varied from approximately 10° A. T. C. at 600 r. p. m. to 2° A. T. C. at 1,800 r. p. m. The ignition lag measured in seconds was reduced one-half by an increase in engine speed of from 600 to 1,200 r. p. m., primarily because more heat was brought to the fuel by the higher air-flow speed.

CONNECTING-PASSAGE DIAMETER

In order to investigate the effect of connecting-passage diameter, the 50-percent prechamber was selected and the diameter of the connecting passage was varied. As in former tests, the single passage used was

circular in cross section. This shape was retained because, with a circular passage, there is a minimum change in clearance shape as the cross-sectional area of the passage is increased. A connecting passage considered too small for practical operation was selected and progressively enlarged. (See table III.) The compression ratio varied from 13.2 to 13.7 with change in passage area. Air-flow speeds through the passages of different size used were calculated by the method given in reference 5 and the results are shown in figure 9.

The general operating and combustion characteristics of the engine changed as the diameter of the connecting passage was varied and the data recorded during the tests are shown in table III.

TABLE III

GENERAL OPERATING CHARACTERISTICS—PASSAGE DIAMETER

[Engine speed, 1,500 r. p. m.; fuel consumption, 3.0×10^{-4} lb./cycle; 12 percent excess air; cylinder head as shown in fig. 3 (a); compression ratio, 13.5; N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 2.5.]

Passage diameter, in.	Passage area, sq. in.	Idling	Injection range (allowable knock to miss), Crank angle, degrees	Cyclic variation of maximum explosion pressure		Combustion sound intensity and regularity	Carbon deposits	
				Chamber, lb./sq. in.	Cylinder, lb./sq. in.		Chamber	Cylinder
3/16 [1]	0.028	Good	38 B. T. C. to 2 A. T. C.		Small	Quiet, regular	None	Light.
17/64 [1]	.055	do	18 B. T. C. to 2 A. T. C.		do	do	do	Do.
3/8	.110	do	11 B. T. C. to 1 A. T. C.		80	Light knock, regular	Increasing carbon.	Increasing carbon.
29/64	.161	do	11 B. T. C. to 1 A. T. C.	100	100	Medium knock, regular	do	Do.
17/32	.222	do	10 B. T. C. to 1 A. T. C.	120	120	Hard knock, regular	do	Do.
21/32	.338	Fair	11 B. T. C. to 1 A. T. C.	100	160	Dull knock, irregular	do	Do.
3/4	.442	Poor	11 B. T. C. to 1 A. T. C.	120	130	do	do	Do.
1 1/16 [2]	.887	Bad	11 B. T. C. to 5 A. T. C.			Light knock, irregular	do	Do.

[1] Erosion of piston crown prevented complete power tests.
[2] Excess heating of exhaust valve and manifold prevented complete tests.

Motoring characteristics shown in figure 10 indicate that friction increases rapidly when passages of less than 29/64-inch diameter are used. The large effect on friction mean effective pressure in this range is due mostly to passage throttling losses because the mechanical and induction losses remain nearly constant (reference 6). The pressure difference between chamber and cylinder is greater with the smaller passages than with the larger ones, which approach the integral combustion chamber condition and show little pressure difference. Figure 11 is a representative motoring card and shows the lag of chamber pressure behind cylinder pressure. The effect of speed on compression pressures and friction mean effective pressure is shown in figure 12. These curves illustrate the increasing

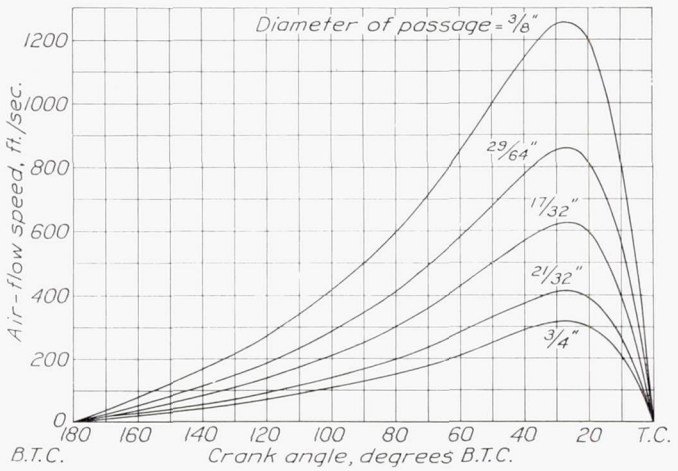


FIGURE 9.—Relationship of air-flow speed to crank position and passage diameter during the compression stroke of a 5- by 7-inch engine with a 12-inch connecting rod. Engine speed, 1,500 r. p. m.; cylinder head as shown in figure 3 (a); 50-percent chamber.

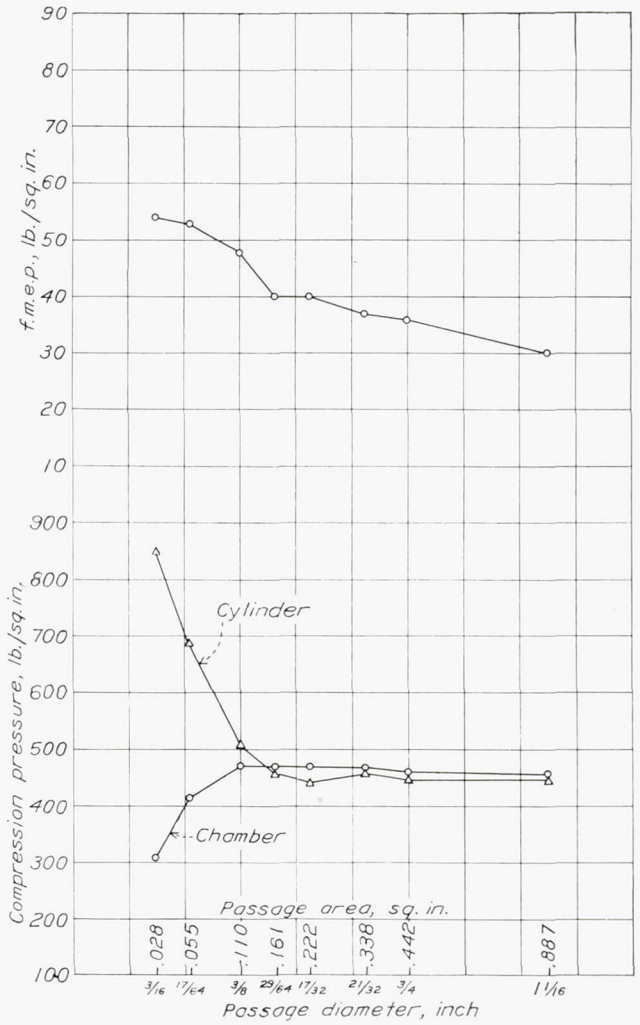


FIGURE 10.—Effect of passage diameter on motoring characteristics. Engine speed, 1,500 r. p. m.; cylinder head as shown in figure 3 (a); compression ratio, 13.5; 50 percent chamber.

effect of the passage area on the different variables as the engine speed is increased.

Figure 13 shows that, for the clearance shape used in these tests, a connecting passage of approximately $^{29}/_{64}$ inch in diameter will give nearly optimum performance over the speed range investigated. Because the airflow velocity through the passage depends on engine speed, the consistent performance over a wide speed range indicates that the longer time available for the preparation of the mixture at low speeds compensates for the lower velocity of the air through the passage and makes good performance with satisfactory engine-operating conditions possible over a wide speed range. In this instance, the criterions for satisfactory engine-operating conditions are moderate cylinder pressures, rates of pressure rise, and combustion sound. At 1,500 r. p. m., while the smallest passage was on test, the pressure in the chamber could not be measured because the engine-operating conditions caused the repeated failure of the pressure-measuring apparatus.

Although the combustion is evidently better at high speed and with small passage diameters, the performance is not the optimum owing to the excessive throttling of the small passages. Throttling becomes less important with decrease in engine speed and the resulting performance curves at 1,000 and 500 r. p. m. are quite flat. In the design of a prechamber, this lack of sensitivity at low engine speeds is therefore advantageous because an optimum passage size for the maximum engine speed can be selected and the performance at lower speeds will not be adversely affected.

Figure 13 shows at 1,500 r. p. m. an increase in ignition lag and a decrease in the rate of pressure rise in both chamber and cylinder as the passage diameter is increased. Increase in ignition lag in the tests of passage sizes ranging from $^{3}/_{8}$-inch to $^{17}/_{32}$-inch diameter was accompanied by an increase in combustion knock; however, for the two larger passages the ignition lag increased slightly, but the combustion knock became less intense. (See knock rating of table III.) In the opinion of some investigators, combustion knock is caused by a high rate of pressure rise. The results of these tests indicate that this condition is not always true because, at a speed of 1,500 r. p. m., the passage giving the highest rate of pressure rise gave the quietest engine operation.

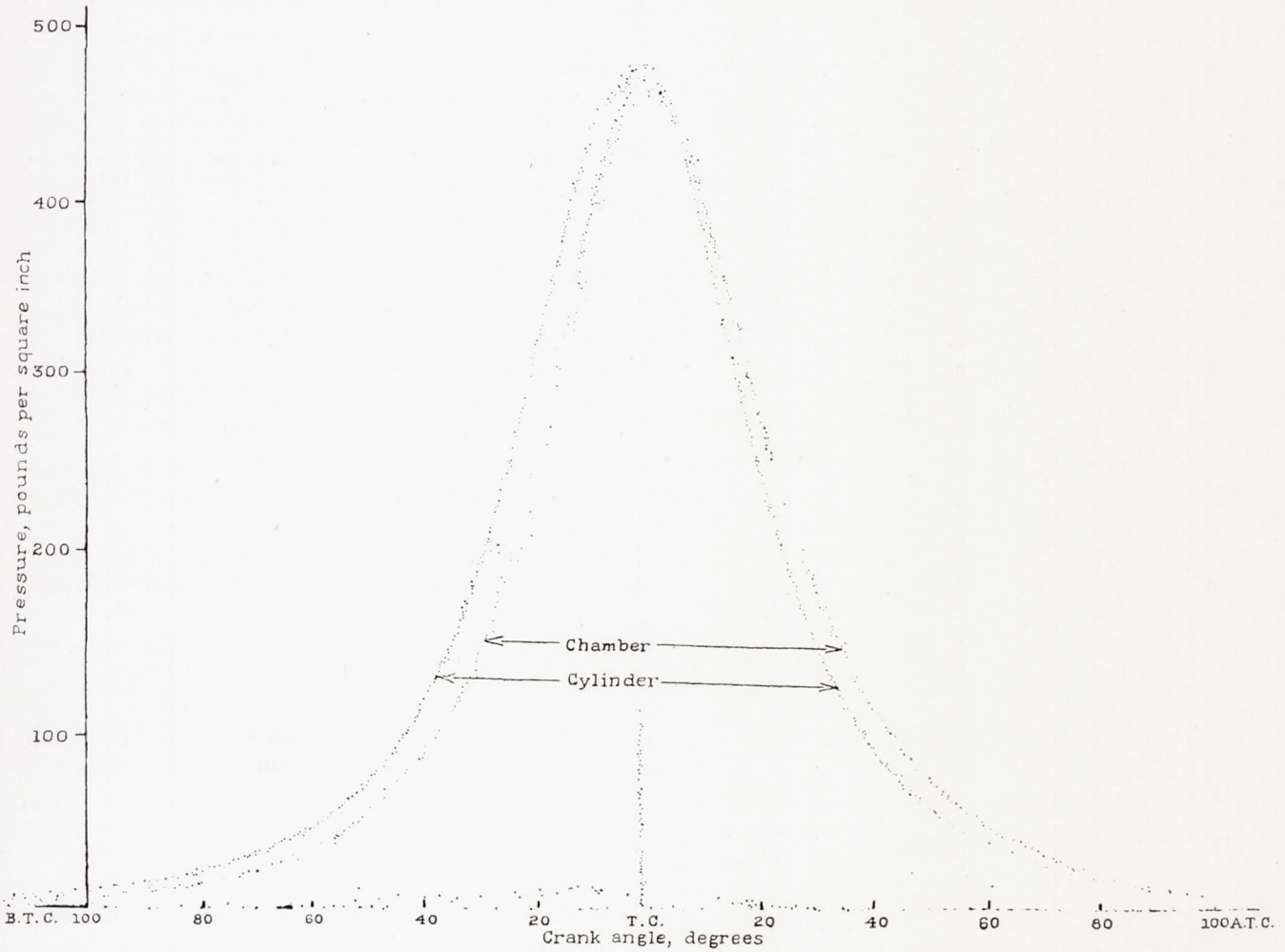


FIGURE 11.—Motoring indicator card. Engine speed, 1,500 r. p. m.; cylinder head as shown in figure 3 (a); passage diameter, 29/64 inch; compression ratio, 13.5.

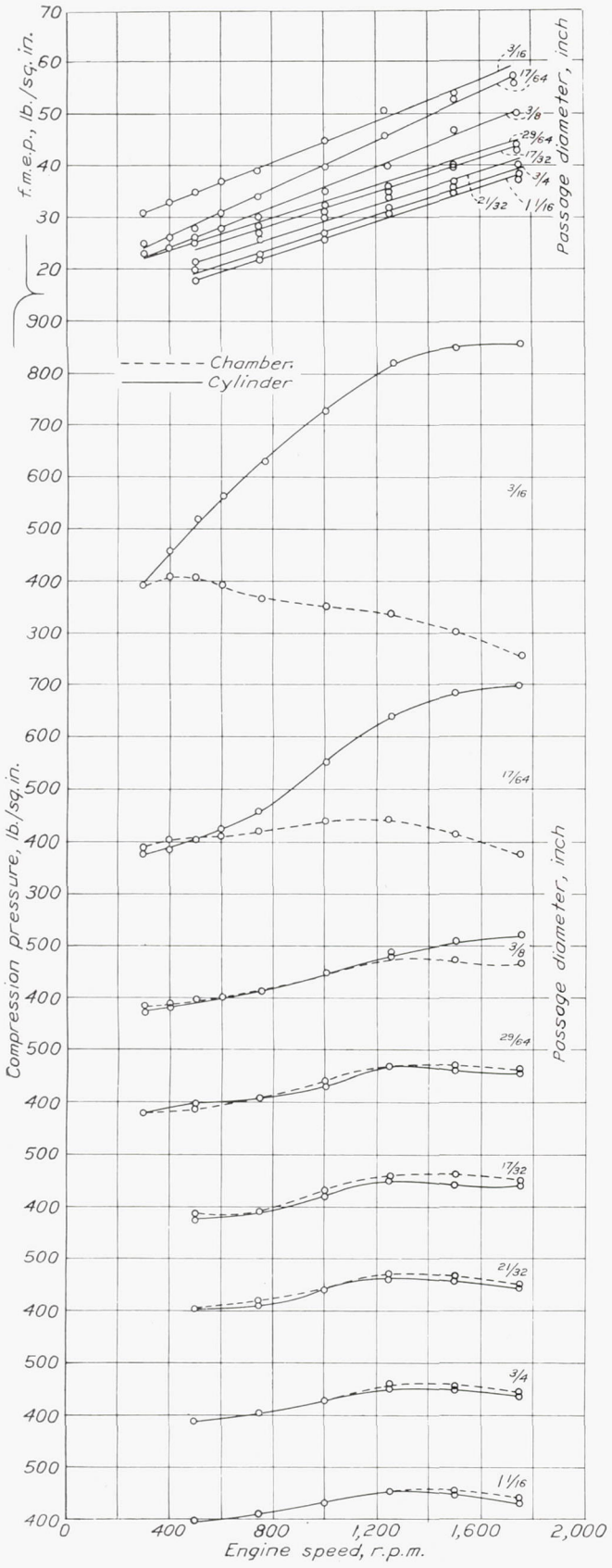


FIGURE 12.—Effect of speed on compression pressures and f. m. e. p. Cylinder head as shown in figure 3 (a); compression ratio, 13.5.

A conclusion drawn from the results of these tests is that combustion knock is more dependent upon ignition lag than upon rate of pressure rise; however, the effect of a small change in either condition is not consistent. The tests made at 1,000 and 500 r. p. m. (fig. 13), owing to the lesser velocities of air flow at these speeds, do not show trends as sharply defined as those shown at 1,500 r. p. m. At each speed, the injection advance angle and rate of fuel injection were held constant for the series of passage diameters tested.

The curves show that some combustion-pressure control can be obtained by means of small passage diameters because with the ⅜-inch-diameter passage the rates of pressure rise are higher in the chamber than in the cylinder. The equivalent data could not be obtained from either of the two smaller passages because, after short power runs, the piston crown was dangerously eroded by the impingement of the concentrated jet of burning gases issuing from the small passage. Small passages, however, do give good mixture control and minimize the effects of irregularities of the fuel-injection system, such as small variations in the start of injection. This effect is shown by the small cyclic variations in cylinder explosion pressure as measured with the balanced-diaphragm pressure indicator. The combustion obtained using the largest passage tested was so slow that the exhaust valve and exhaust manifold became red hot after a few minutes of operation.

Supplementary tests made at maximum allowable advance angle are represented in figure 13 by the points that do not fall on the curves. These runs were made because it was found that the explosion pressures were decreasing with an increase in passage diameter and it was considered advisable to determine whether the best performance could be equaled by advancing the injection and thereby raising the explosion pressures. The results of these tests at maximum allowable advance angle show that, although the maximum explosion pressures were considerably increased, the performance was only slightly improved. The combustion knock under these conditions was much worse than when testing any passage and using optimum injection advance angle.

COMBUSTION-CHAMBER SHAPE

Clearance distribution and connecting-passage diameter were considered the most important variables in the design of a prechamber cylinder head and therefore they were extensively investigated. Several lesser variables that contribute to the performance characteristics of the combustion chamber were also investigated.

The prechamber was kept at 50 percent of the total clearance for most of the tests, and the connecting passage was maintained at 9/16-inch diameter or the equivalent area. Although these proportions are not

the optimum for prechamber design, the sacrifice in performance was sufficiently small to justify their use to maintain continuity throughout the entire investigation.

The passage was brought into the chamber radially and tangentially (fig. 3 (b)) by using inserts designed and constructed to permit such variations. When the tangential passage was used, the direction of the passage to the cylinder was changed by rotating the chamber cap and passage insert as a unit into positions as far as 72° to the right and to the left. The ends of the passage were successively flared to determine the effect of passage flaring.

The effect of prechamber shape was investigated for a limited series of tests. Analysis and test results indicated the advisability of confining the test shapes to volumes of revolution in order to conserve the residual air flow within the chamber. The spherical chamber of the first tests was changed to a disk rounded at the outer edge and arranged vertically so that the plane of the disk was parallel to the axis of the engine cylinder. The connecting passage was introduced tangentially to the disk (fig. 3(c)). Three injection-valve locations were provided as shown, and power tests were made with the fuel valve in each.

The effect of increasing the quantity of air rotated in the prechamber was investigated by changing the volume of a spherical chamber from 50 to 70 percent of the clearance volume. The tangential passage was substituted for the radial passage and comparable tests were made. The approximate direction of the air flow for both the 50- and the 70-percent chambers with radial and tangential passages was indicated by air-flow patterns made by extending a number of copper nibs into the auxiliary chamber from a gasket clamped between the two parts of the chamber, as described in reference 4. In order to take the air-flow patterns, the engine was started from rest, motored up to 1,500 r. p. m. as quickly as possible, and then stopped. The variation in performance when the engine was operated with the fuel valve first in the central and then in the top injection-valve location was determined for both prechambers.

The general operating characteristics of the engine—that is, starting and idling ability, cyclic regularity, and combustion shock—were little affected by any of the changes made during these tests. A change from the Diesel fuel used in previous tests to Auto Diesel fuel greatly reduced the combustion knock, increased the injection-advance-angle range, and decreased the cyclic variation in maximum cylinder pressure from ±75 to ±40 pounds per square inch. Some combustion knock was present in all tests but was not considered serious. The rates of pressure rise in the cylinder and the prechamber were, respectively, 68 and

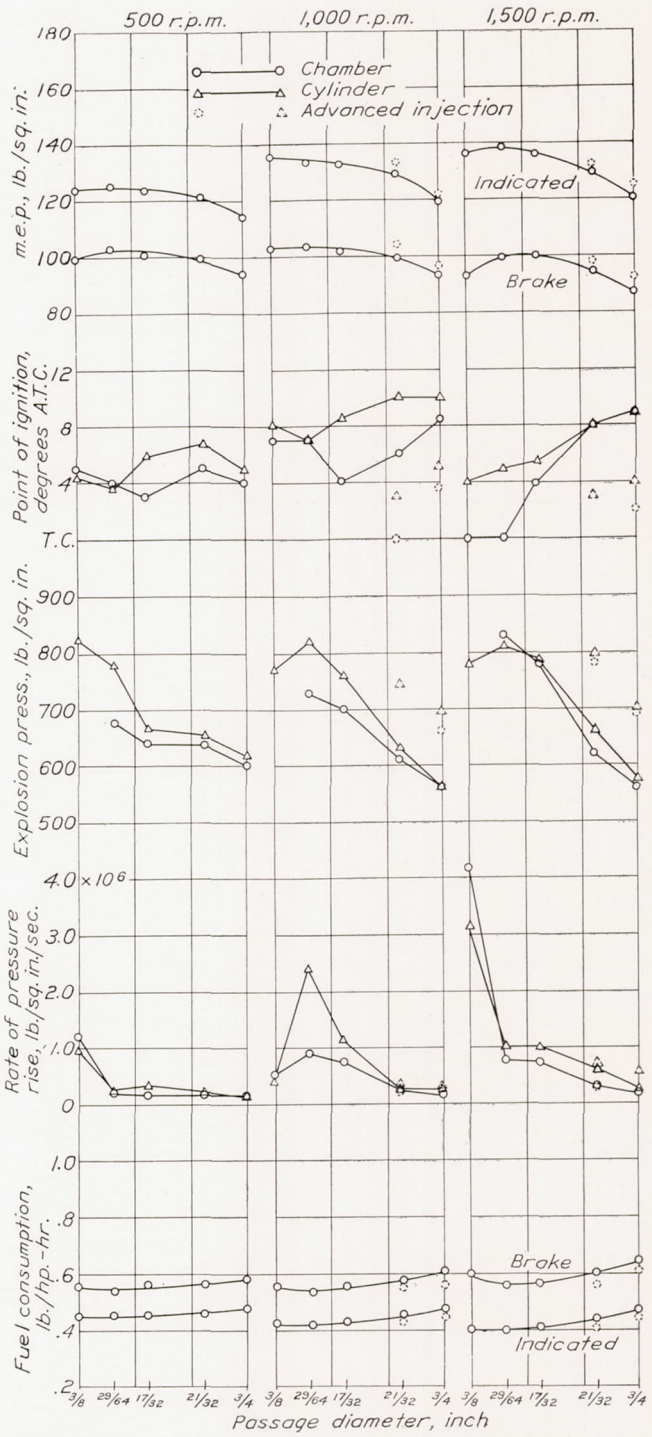


FIGURE 13.—Effect of passage diameter on engine performance. Fuel consumption, 3.25×10^{-4} lb./cycle; no excess air; cylinder head as shown in figure 3 (a); compression ratio, 13.5; N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 2.5; injection advance angle, 11° B. T. C.

45 pounds per square inch per degree at 1,500 r. p. m. with the best combination of variables covered in this report. Rates of pressure rise for previous work, in which the spherical chamber and the original fuel were used, were in the order of 85 and 75 pounds per square

inch per degree at 1,500 r. p. m. for cylinder and chamber, respectively.

The purpose of directing the passage tangentially to the chamber instead of radially was to create a high-velocity, rotational, residual air flow in the auxiliary chamber to improve the fuel and air mixing. Table IV shows the effects of flaring and of changes in the intensity of the rotational air swirls obtained by several combinations of connecting passage. Flaring the passage had very little effect on the performance. Several other methods of spreading the gases over the piston crown were tried, such as an elliptical passage insert and a three-passage insert designed according to the proportional-orifice principle (reference 7). The effect of these changes on the performance was also negligible.

As a rotational swirl was found to be effective in the chamber, it was decided to determine the effect of a swirl in the cylinder. This effect was produced by changing the passage direction in the cylinder. Carbon deposits showed that a swirl was produced, but there was no appreciable improvement in engine performance.

TABLE IV

EFFECT OF PASSAGE DIRECTION AND FLARE ON ENGINE PERFORMANCE

[Fuel consumption, 3.25 $\times 10^{-4}$ lb./cycle; no excess air; cylinder head as shown in fig. 3 (b); compression ratio, 13.5; N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 2.5]

	1,000 r. p. m.					1,500 r. p. m.				
	i. m. e. p., lb./sq. in.	b. m. e. p., lb./sq.in.	Explosion pressure, lb./sq.in.	Fuel consumption		i. m. e. p., lb./sq. in.	b. m. e. p., lb./sq. in.	Explosion pressure, lb./sq. in.	Fuel consumption	
				lb/.i.hp.-hr.	lb./b.hp.-hr.				lb./i.hp.-hr.	lb./b.hp.-hr.
RADIAL PASSAGE										
Slight rotational air flow	137	109	795	0.42	0.52	133	98	750	0.42	0.58
Strong rotational air flow	139	111	740	.41	.51	137	102	730	.41	.55
TANGENTIAL PASSAGE										
Strong rotational air flow	141	112	740	0.38	0 48	142	104	720	0.38	0.53
Cylinder end of passage flared	141	112	740	.39	.49	144	106	730	.38	.52
Both ends of passage flared	140	113	800	.39	.49	141	105	770	.40	.54

Table V shows the effect of changing the prechamber shape from a sphere to a disk of equal volume and also the effect of fuel-valve location in the disk chamber. Tangential passages were used in both cases. The disk chamber with tbe injection valve in the central location gave an improvement in i. m. e. p. at 1,500 r. p. m. over the spherical chamber under similar conditions, which can be attributed to the fact that in the disk chamber the low-velocity zones of the spherical chamber were removed and the rotating mass of air was in the zone of the single fuel spray and the connecting passage. This relation of chamber shape and fuel spray evidently resulted in better mixing in the prechamber with the resultant improved performance.

TABLE V

EFFECT OF PRECHAMBER SHAPE AND FUEL-VALVE LOCATION ON ENGINE PERFORMANCE

[Fuel consumption, 3.25$\times 10^{-4}$ lb./cycle; no excess air; cylinder head as shown in figs. 3 (b) and 3 (c); compression ratio, 13.5; N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 2.5]

Fuel-valve location	1,000 r. p. m.										1,500 r. p. m.									
	i. m. e. p., lb./sq. in.		b. m. e. p., lb./sq. in.		Explosion pressure, lb./sq. in.		Fuel consumption				i. m. e. p., lb./sq. in.		b. m. e. p., lb./sq. in.		Explosion pressure, lb./sq. in.		Fuel consumption			
							lb./i. hp.-hr.		lb./b. hp.-hr.								lb./i. hp.-hr.		lb./b. hp.-hr.	
	Flame start	Full load	Flame start	Full load	Flame start	Full load	Flame start	Full load	Flame start	Full load	Flame start	Full load	Flame start	Full load	Flame start	Full load	Flame start	Full load	Flame start	Full load
SPHERICAL CHAMBER (FIG. 3 (b))																				
2	130	140	103	113	800	790	0.36	0.39	0.46	0.48	126	140	91	105	740	760	0.33	0.40	0.45	0.54
DISK CHAMBER (FIG. 3 (c))																				
3	99	124	72	98	780	790	0.40	0.44	0.55	0.56	101	128	66	92	780	760	0.37	0.45	0.57	0.62
2	138	141	111	113	780	780	.38	.41	.47	.50	133	145	98	109	740	740	.35	.38	.48	.51
1	143	147	115	119	780	760	.37	.38	.47	.48	143	149	106	113	740	740	.36	.38	.48	.51

The effect of fuel-valve location is also shown in table V; the great difference in maximum power for the three fuel-valve locations provided in the disk chamber indicates the importance of the position of the fuel spray relative to the air movement. The best performance was obtained with the spray axis from the single-orifice nozzle directed in the same plane with the air flow and at only a small angle from the direction at right angles to the air flow. The spray was also directed toward the connecting passage. The worst performance was obtained with the fuel valve in the lowest position (see fig. 3 (c)) in which the spray was injected counter to the air flow and, with the injection timing used, should have penetrated directly through the passage to the cylinder. The fact that the fuel did penetrate at least to the passage was indicated by carbon formation around the mouth of the passage. This arrangement was made to obtain a rich mixture adjacent to the passage ready to be ejected into the cylinder by the pressure resulting from the combustion in the chamber. The chamber cap was rotated 180° so that the lower valve position was in approximately the same location but the spray was directed, not through the passage, but above the entrance to the passage and at an angle to the air flow, not counter to it. This condition increased the brake mean effective pressure approximately 10 pounds per square inch over that originally obtained with the fuel valve in the lowest position.

An analysis of previous work indicated that the greater the amount of air in motion the better would be the mixing of the fuel and air and, consequently, the better the performance. A tangential passage in conjunction with a chamber that contained a larger percentage of the clearance volume was used to increase the quantity of air in motion. Table VI shows that this analysis was correct for spherical chambers because, with the fuel valve in the central location, the improvement in the performance with increase in chamber volume was greater when a tangential passage was used than when a radial passage was used. Increase in rotational, and probably residual, air-flow velocity due to the tangential passage was sufficient to make an appreciable difference in the performance. The investigation was made with spherical chambers although the maximum performance would be less than with the disk chambers; the indicated trend, however, should be the same for both auxiliary-chamber shapes.

The tangential connecting passage was used because introducing the air tangentially to a volume of revolution assisted in setting up a rotational swirl in the chamber, which should persist after the piston had reached the upper limit of its travel. As this residual air flow was believed to be the cause of the increased power, every attempt was made to intensify and preserve the flow. This theory could not be definitely proved because there are no means available for measuring the velocity of the flow; the predominating direction of the air flow, however, was determined by means of the air-flow patterns. The radial passage to the same chamber also showed rotational air flow but of less intensity and in the opposite direction. This condition was probably caused by the short passage used, which permitted some air from the cylinder to pass directly into the prechamber without being directed by the passage. (See fig. 3 (b).) The passage was as long as the construction of the head would permit.

TABLE VI

EFFECT OF PASSAGE DIRECTION, PRECHAMBER VOLUME, AND FUEL-VALVE LOCATION ON ENGINE PERFORMANCE

Engine speed, 1,500 r. p. m.; fuel consumption, 3.25×10^{-4} lb./cycle; no excess air; cylinder head as shown in fig. 3 (b); compression ratio, 13.5; N. A. C. A. 7A pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 2.5]

Passage	i. m. e. p., lb./sq. in.		b. m. e. p., lb./sq. in.		Explosion pressure, lb./sq. in.		Fuel consumption			
							lb./i. hp.-hr.		lb./b. hp.-hr.	
	Flame start	Full load	Flame start	Full load	Flame start	Full load	Flame start	Full load	Flame start	Full load
FUEL-VALVE LOCATION 2										
50-percent radial	133	138	98	103	730	720	0.35	0.41	0.48	0.56
50-percent tangential	138	141	103	106	760	760	.33	.40	.46	.52
70-percent radial	133	140	98	104	830	830	.34	.41	.48	.55
70-percent tangential	139	144	104	109	800	790	.36	.38	.48	.51
FUEL-VALVE LOCATION 1										
50-percent radial	131	141	99	109	790	800	0.35	0.41	0.48	0.52
50-percent tangential	131	135	98	102	820	820	.38	.42	.51	.55
70-percent radial	141	147	108	112	880	880	.35	.39	.47	.52
70-percent tangential	132	138	98	104	800	780	.36	.42	.49	.55

As the air flow with either passage is rotational, the differences in performance shown for the different fuel-valve locations (table VI) are more readily understood. It was found that with either passage in the spherical chamber the performance was improved by injecting the fuel at the point "upstream" on the circumference of the chamber. Location 1 was better when using the radial passage and location 2 better when using the tangential passage. (See fig. 3 (b).) In the disk chamber, injecting the fuel near the passage mouth but directly toward the passage gave the worst results.

CYLINDER-CLEARANCE SHAPE

In the tests of clearance distribution and passage size, a domed piston crown was employed. At the conclusion of these tests, the performance with the domed crown was compared with that using first a flat crown, second a dished crown, and third a piston crown with all the cylinder clearance concentrated in front of the connecting passage. Extensive tests were made using this latter type of piston crown, but the performance was inferior to either of the other two piston crowns.

For the sake of convenience in testing, subsequent tests were made using a flat piston crown since there was little difference between the performance of the flat and the domed crown piston.

INJECTION SYSTEMS

The 50-percent disk chamber with the tangential passage was assembled with head 7 at a compression ratio of 13.5 and, using fuel-valve location 1, the injection system was varied to determine the effect on engine performance. Three fuel pumps having widely different characteristics were selected and tested with different fuel-valve assemblies that gave a variety of injection systems. The N. A. C. A. 7A and the commercial fuel pump used were cam-operated and of constant stroke but had different rates of displacement. The third injection pump was the N. A. C. A. 12, a cam-operated plunger type, but one in which injection is caused by the release of pressure stored in a reservoir of correct volume. This pump in combination with the correct valve gives a fast rate of injection and was used to obtain a shorter injection period than that of any of the other pump and valve combinations. The N. A. C. A. 7A fuel pump and the single 0.050-inch-diameter orifice were used to determine the effect of the orifice length-diameter ratio. Results obtained by using multiple orifices to distribute the fuel by injection as well as by air flow were also determined. Tests of a pintle nozzle with two different fuel pumps were included in the injection-system investigation. With the injection system that gave the best engine performance, tests were made to determine the effect of injection advance angle and fuel quantity on engine performance.

In order to improve the inherently poor starting characteristics of this type of combustion chamber, a series of starting tests was made, injecting the fuel directly into the cylinder instead of into the prechamber. An 0.008-inch-diameter orifice nozzle was used with a valve-opening pressure of 2,500 pounds per square inch. The size of the orifice used limited the fuel injected to very small quantities. The engine was motored at gradually increasing speeds until the engine started firing and the speed could be maintained under its own power. For comparison, this procedure was repeated with the same fuel nozzle in the prechamber.

Table VII shows the results of the injection-system tests. The object of these tests was to obtain a fuel spray the characteristics of which best suited the 50-percent vertical-disk chamber and tangential passage, the arrangement that had given the best performance for this size of chamber. Injection period was the only spray characteristic accurately measured because in this type of chamber with its high air-flow speed other characteristics, such as distribution of fuel within the spray and spray cone angle, should not be critical. As shown in the table the optimum performance based on the i. m. e. p. at flame start and full load was obtained using the N. A. C. A. 7A fuel pump, 13A fuel valve, and 0.050-inch nozzle with a length-diameter ratio of 6. Multiple-orifice nozzles were tried but, as they were definitely inferior, the performance is not included.

TABLE VII

EFFECT OF INJECTION SYSTEMS ON ENGINE PERFORMANCE

[Engine speed, 1,500 r. p. m.; fuel consumption, 3.25×10^{-4} lb./cycle; no excess air; cylinder head as shown in fig. 3 (c); compression ratio, 13.5; fuel valve in location 1]

Injection system	i. m. e. p. lb./sq. in.		b. m. e. p. lb./sq. in.		Explosion pressure lb./sq. in.		Fuel consumption				Relation of injection to top center, crank angle, degrees
							lb./i. hp.-hr.		lb./b. hp.-hr.		
	Flame start	Full load	Flame start	Full load	Flame start	Full load	Flame start	Full load	Flame start	Full load	
N. A. C. A. 7A pump:											
13A valve:											
0.050-inch-diameter orifice:											
Length-diameter ratio, 2.5	145	149	109	112	700	700	0.35	0.38	0.48	0.49	7 B. T. C. to 14 A. T. C.
Length-diameter ratio, 4.0	143	148	107	111	710	700	.36	.38	.48	.50	Do.
Length-diameter ratio, 5.0	145	149	108	112	.....	.....	.35	.37	.47	.50	Do.
Length-diameter ratio, 6.0	146	150	109	113	700	700	.35	.37	.48	.49	Do.
0.040-inch-diameter orifice	132	142	95	106	640	630	.35	.40	.49	.53	7 B. T. C. to 15 A. T. C.
0.070-inch-diameter orifice	145	148	108	112	740	730	.36	.38	.48	.49	7 B. T. C. to 12 A. T. C.
Commercial valve; pintle nozzle	141	147	104	110	720	720	.35	.36	.47	.50	7 B. T. C. to 16 A. T. C.
Commercial pump:											
13A valve; 0.050-inch-diameter orifice; 3,500 lb./sq. in. valve-opening pressure.	140	150	103	113	760	740	.33	.38	.46	.50	7 B. T. C. to 14 A. T. C.
Commercial valve; pintle nozzle, 3,500 lb./sq. in. valve-opening pressure.	133	143	98	108	740	790	.35	.39	.48	.51	4 B. T. C. to 20 A. T. C.
N. A. C. A. 12 pump:											
13A valve; 0.050-inch-diameter orifice; 3,500 lb./sq. in. valve-opening pressure.	141	147	104	111	860	870	.34	.38	.47	.50	7 B. T. C. to 27 A. T. C.
17—$\frac{3}{16}$ F. I. S. valve; 0.059-inch-diameter orifice; 6,000 lb./sq. in. valve-opening pressure.	128	144	92	108	720	720	.35	.39	.48	.52	3 B. T. C. to 7 A. T. C.
17—$\frac{3}{16}$ F. I. S. valve; 0.051-inch-diameter orifice, 3,500 lb./sq. in. valve-opening pressure.	.....	150	.....	112	.....	730	.....	.38	.....	.50	4 B. T. C. to 11 A. T. C.

Marked improvement in the capability of the engine to start at a temperature of about 70° F. was obtained by injecting the fuel into the cylinder. By the use of a nozzle with a single 0.008-inch-diameter orifice and the injection of only a small percentage of full-load fuel quantity, the engine could be started by motoring the engine at from 200 to 300 r. p. m.; whereas, when the fuel was injected into the chamber with the same nozzle, a speed of 600 to 700 r. p. m. was required. Improvement in starting is due to a higher temperature in the cylinder.

Figure 14 shows the results of the variable-injection-advance-angle tests. These curves are characteristic

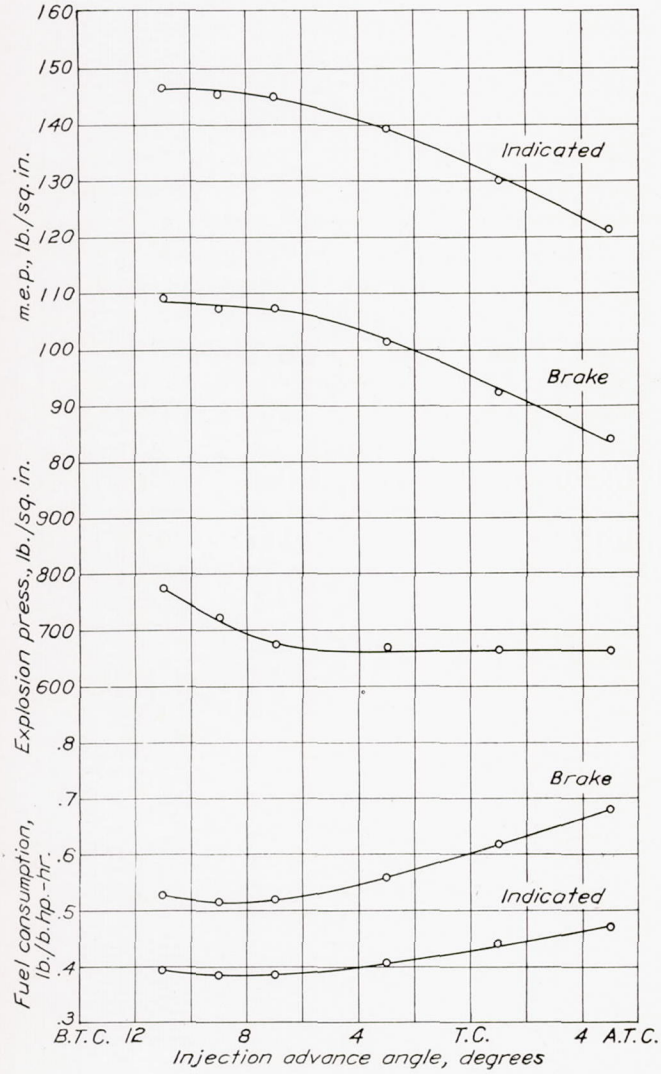


FIGURE 14.—Effect of injection advance angle on engine performance. Engine speed, 1,500 r. p. m.; fuel consumption, 3.0×10^{-4} lb./cycle; 12 percent excess air; cylinder head as shown in figure 3 (c); compression ratio, 13.5; N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 2.5.

of the prechamber type of combustion chamber. For these tests the optimum advance angle was 7° before top center because at this point increase in mean effective pressure stopped while increase in maximum cylinder pressure began.

The results of the variable-fuel-quantity test are shown in figure 15. The curve of mean effective pressure against fuel quantity shows the characteristic straight line at small fuel quantities but, for the combustion chamber under test, the curve continues straight to comparatively large fuel quantities. The mean

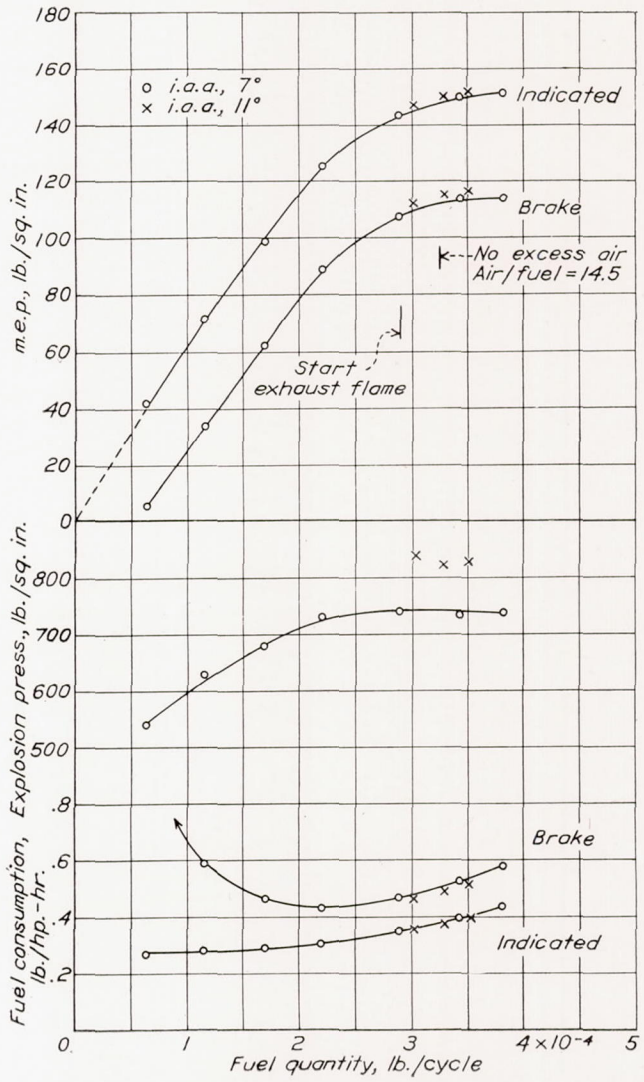
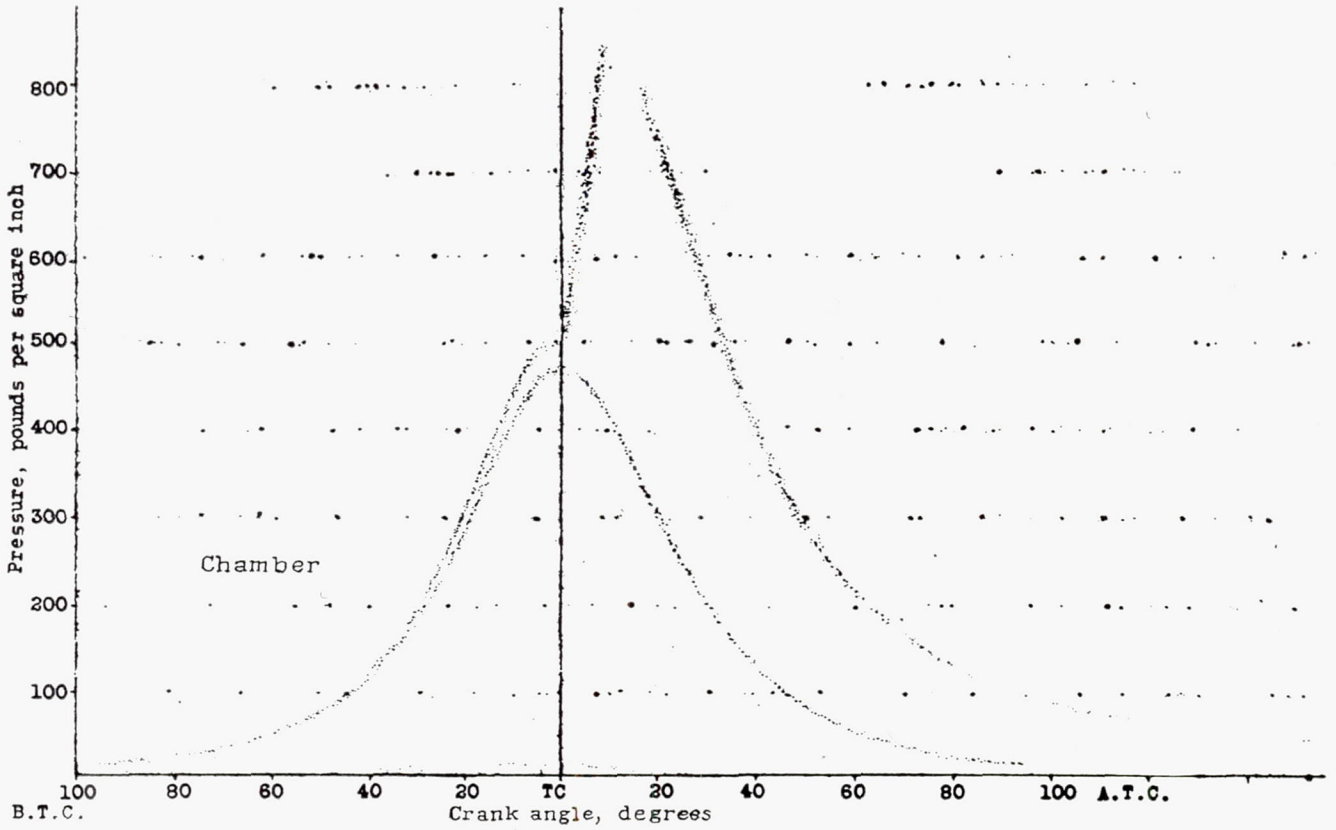


FIGURE 15.—Effect of fuel quantity on engine performance. Engine speed, 1,500 r. p. m.; cylinder head as shown in figure 3 (c); compression ratio, 13.5; N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 6.

effective pressure varies linearly with the quantity of fuel injected up to a fuel quantity of 2.25×10^{-4} pound per cycle (air-fuel ratio approximately 23) but the curve begins to droop at this point; when the fuel quantity is increased to 2.90×10^{-4}, flame appears in the exhaust. With this type of combustion chamber, flame appears in the exhaust before smoke. Both flame and smoke can be seen in the exhaust at full-load fuel quantity.

The points of figure 15 that do not fall on the curve represent the data obtained at a 4° increase of the injection advance angle. It will be noted that the explosion pressure increased out of proportion to the increase in engine performance, and therefore the



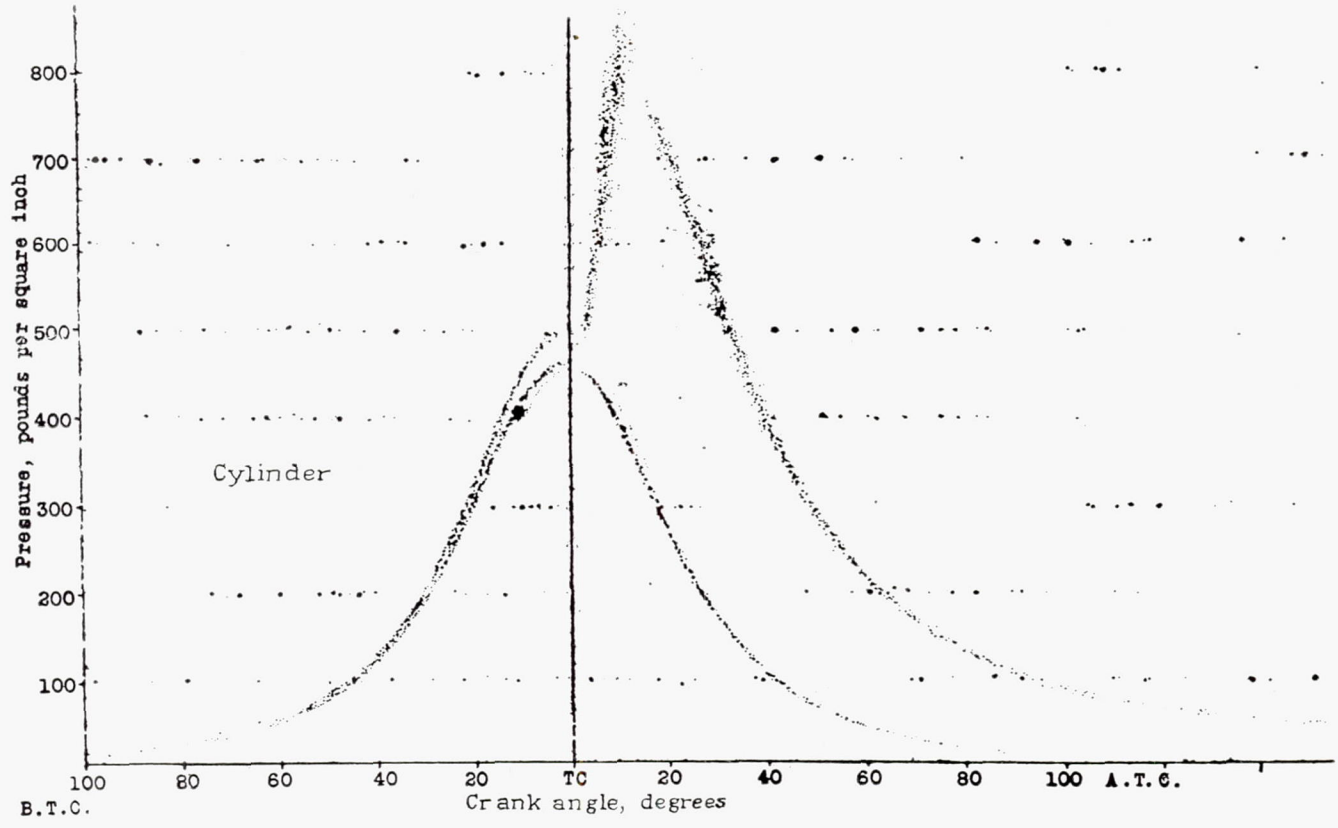


FIGURE 16.—Typical power and motoring indicator cards. Engine speed, 1,500 r. p. m.; fuel consumption, 3.25×10^{-4} lb./cycle; no excess air; cylinder head as shown in figure 3 (c); N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 6; injection advance angle, 7° B. T. C.

injection advance angle was not further increased. In all except these tests the injection advance angle was determined by the combustion sound. In these tests, although the sound of combustion became more intense, the condition was not considered dangerous.

At the full-load fuel quantity during the variable-load run, made with the best combustion-chamber shape and fuel-injection system, indicator cards typical of those obtained from this engine were taken from the chamber and cylinder (fig. 16). The rates of pressure rise determined from the indicator diagrams are 68 and 45 pounds per square inch per degree for the cylinder and prechamber, respectively.

COMPRESSION RATIO

The lack of engine data concerning the influence of compression ratio on engine operation and performance made an investigation of compression ratio desirable (reference 8). An attempt was first made to follow the usual procedure of changing only one variable at a time. After some preliminary tests, however, this procedure was found to be so expensive that it was decided to permit variation of the clearance distribution and to change the compression ratio by raising or lowering the head, i. e., by varying the cylinder-clearance volume. Explosion pressures were kept nearly constant by controlling the injection advance angle.

The results of the tests at different compression ratios are shown in figure 17. The curves have been plotted without correcting for the change in performance due to the increase in relative chamber size, which would amount to about one-half the mean-effective-pressure increase shown at the highest compression ratio. When the chamber size is taken into account, there is still a small but definite trend toward an increase in indicated power with an increase in compression ratio. This trend is in agreement with theoretical analysis, which indicates higher cycle efficiencies at higher compression ratios. The brake performance shows very little change, possibly because the increased cycle efficiency at the higher compression ratios was counteracted by the decreased mechanical efficiency.

Starting and general operating characteristics improved with increasing compression ratio. At the highest compression ratio, the increased compression temperature reduced the ignition lag and caused the combustion knock to soften and practically disappear. As the compression ratio was increased, it was found necessary to reduce the injection advance angle by several degrees in order to hold the cylinder pressure constant throughout the tests. Limitation of maximum cylinder pressure, however, did not result in a loss of power; in fact, short tests at higher cylinder pressures in some cases showed a slight impairment of performance.

Indicator cards taken at each compression ratio illustrated very clearly the decrease in allowable pressure rise as the compression pressure approached the maximum cylinder pressure; the only gain in performance was from the higher cycle efficiency. A decrease in the rate of pressure rise is also shown on the cards, but otherwise they have the same general shape as the cards heretofore presented; therefore they are not included.

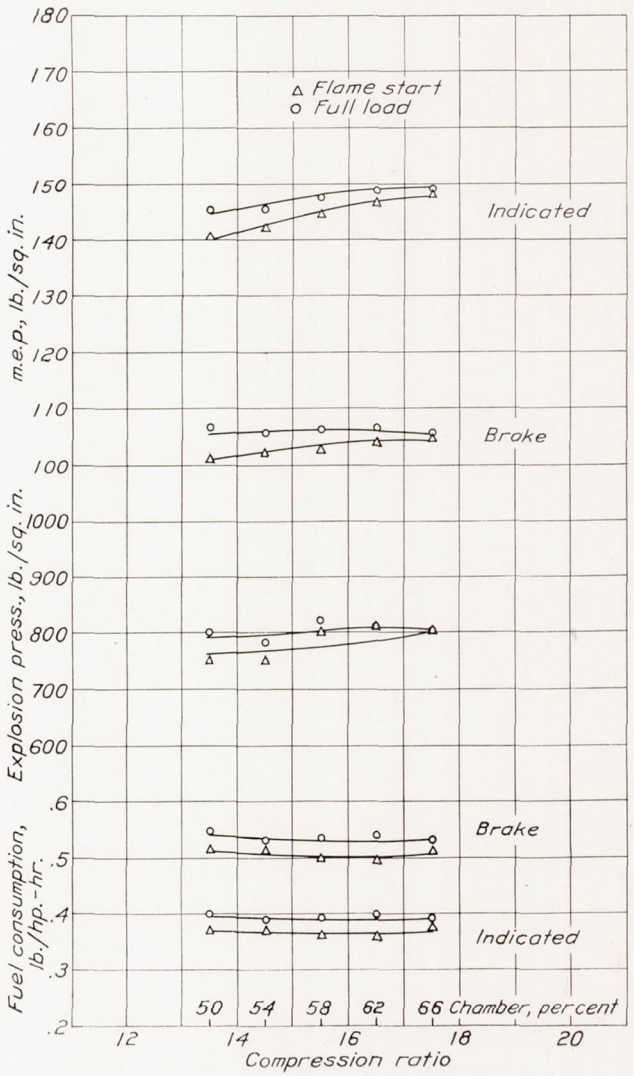


FIGURE 17.—Effect of compression ratio on engine performance. Engine speed, 1,500 r. p. m.; fuel consumption, 3.25×10^{-4} lb./cycle; no excess air; cylinder as shown in figure 3 (c); N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter ratio, 6.

With the disk chamber, the greatest compression ratio that could be obtained was 17.5. The trend of the curves indicates, however, that there would be no improvement in brake performance at higher compression ratios. Although an optimum compression ratio is not clearly defined, the b. m. e. p. at flame start, the easier starting, and the quieter operation favor the use of a high compression ratio in this type of engine.

BOOSTING OF AIR CHARGES

Boosting tests were conducted on the engine assembled with the optimum combination of variables. For continuity throughout the investigation, the compression

ratio was retained at 13.5. A DePalma supercharger with a 12-cubic-foot surge tank was connected to the engine and variable-load runs made at boost pressures

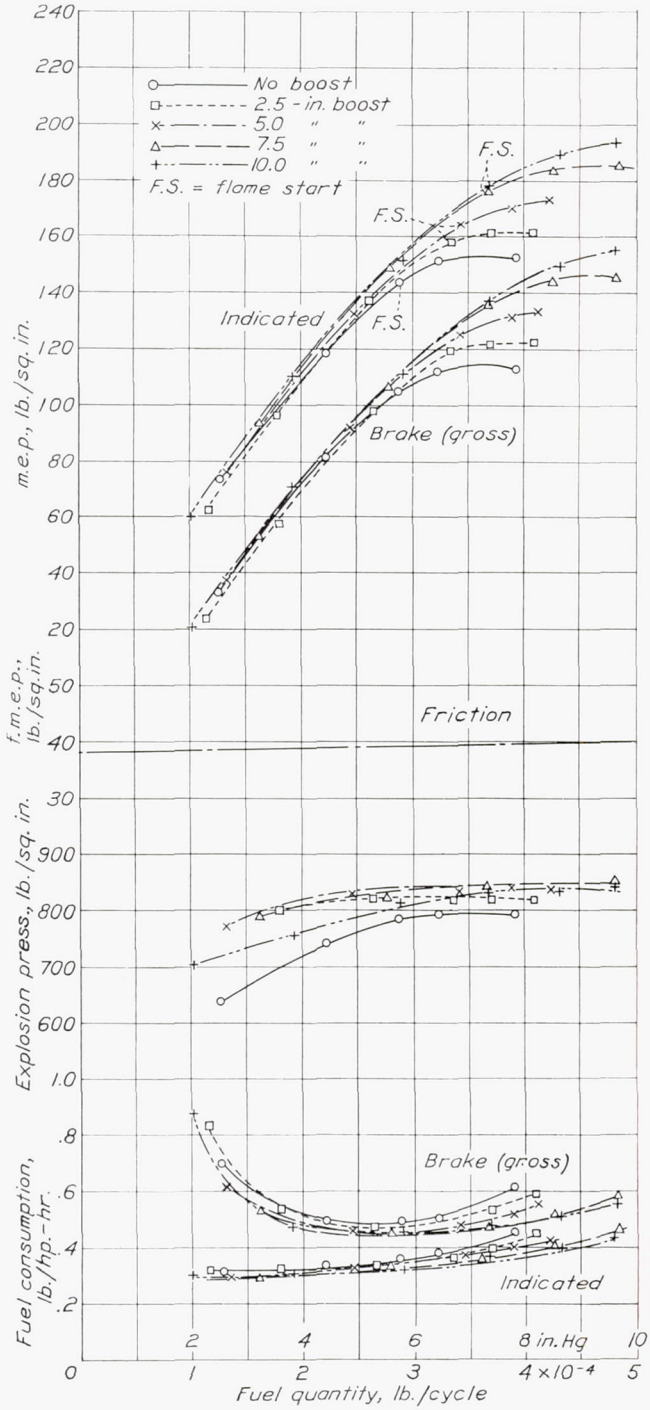


FIGURE 18.—Effect of fuel quantity on engine performance at several boost pressures. Engine speed, 1,500 r. p. m.; cylinder head as shown in figure 3 (c); compression ratio, 13.5; N. A. C. A. 7A fuel pump and 13A fuel valve; 0.050-inch nozzle; length-diameter, 6.

of 0, 2.5, 5, 7.5, and 10 inches of mercury at 900, 1,200, and 1,500 r. p. m. At 1,500 r. p. m. one point was taken at the highest boost pressure obtainable with the supercharger used, about 13½ inches of mercury. Explosion pressures were kept nearly constant by retarding the injection advance angle as the boost pressure was increased. At 1,800 r. p. m. power tests were made at each of the test pressures at the flame-start fuel quantity and at one larger fuel quantity. Friction tests were made by motoring the engine after each power run at the different boost pressures and engine speeds.

The effect of boosting a prechamber type of cylinder head was rather thoroughly investigated because the authors could find no references to previous tests by other investigators. However, only the data at 1,500 r. p. m., which are representative of the entire family of curves, are presented (fig. 18). The m. e. p. curves are quite conventional in shape and, at an air-fuel ratio of 14.5, show a constant increase of from 3 to 4 pounds per square inch per inch of boost pressure.

The objective of holding the explosion pressure constant while the boost pressure was varied during the tests was not exactly accomplished, but the increase in explosion pressure shown was within the limitations of the equipment used. Explosion pressures were controlled by varying the injection advance angle, which was 7° at zero boost pressure and was progressively retarded until at 10 inches of mercury the injection advance angle was only 2°. Since the compression pressure increases with boost pressure and the maximum cylinder pressure was kept constant, the amount of fuel that could be burned at or near top center was decreased by retarding the injection. If it had not been necessary to decrease the injection advance angle, the gain in power with increased boost pressure would have been slightly greater because more fuel could have been burned at or near top center with a resulting greater efficiency. Because of the higher compression pressures with the consequent reduction in ignition lag, however, the power loss was not nearly so great as if the injection advance angle had been reduced the same amount at zero boost pressure.

In view of these facts, it is possible that it might be advantageous to use a compression ratio lower than 13.5 for boosting because the compression pressure would be lower and a greater useful pressure rise could be used. The influence of boost pressure on the combustion shock would permit a lower compression ratio to be used because the combustion shock is diminished with increased inlet pressure, although a lower compression ratio alone would increase it. At the boost pressure of 7.5 inches of mercury, the sound of the engine was very satisfactory at all loads and at each test speed.

An examination of the curves presented herein shows that the friction mean effective pressure, neglecting supercharger friction mean effective pressure, did not decrease with increase in boost pressure. This observation applies for all test speeds. With the integral combustion-chamber type of cylinder head, the friction mean effective pressure, also neglecting the power required by the supercharger, was found to decrease

slightly with increase in boost pressure. This condition did not exist with the type of cylinder head under test, probably because of the increase in pumping loss as the weight of air forced through the connecting passage is increased. At large boost pressures the higher pumping losses more than offset the work done on the piston during the intake stroke, which resulted in a slight increase in friction mean effective pressure.

In these tests the connecting passage was maintained at a fixed diameter, which is probably not the best condition. The ideal way of conducting the tests would have been to determine and use the correct passage size for each boost pressure. The injection period was too long at high boost pressures, as the injection system is designed to deliver, in approximately 20 crankshaft degrees, a fuel quantity of 3.25×10^{-4} pounds per cycle, which is full load at zero boost. The full-load fuel quantity and consequently the injection period, however, increase with boost. At high boost pressures, therefore, the injection period continued too long after top center for efficient combustion. It is believed that the performance at the optimum boost pressure would be improved if the correct passage area and injection period were used; however, the scope of these tests did not include the determination and application of each of these conditions. With this type of combustion chamber improvement in engine performance by scavenging the clearance volume is practically impossible because all the clearance should be in the prechamber away from the valves. Furthermore, owing to lack of mechanical clearance when the piston is on top center, both valves must be closed. This condition is improper for the best exhausting and air charging, which limits the specific output.

CONCLUSIONS

The following specific conclusions are presented:

1. Clearance distribution:

(*a*) For maximum performance the prechamber should be relatively as large as is practicable; however, lower cylinder pressures, less combustion knock, and less heat loss to the cooling water occur with the smaller chamber sizes.

(*b*) The size of the prechamber has a negligible effect on friction mean effective pressure and compression pressures.

(*c*) Variation of clearance distribution only, for a fixed ratio of prechamber volume to connecting passage area, does not sufficiently control combustion or eliminate combustion knock.

2. Connecting-passage diameter:

(*a*) For the engine size and combustion-chamber design used in this investigation, the connecting-passage diameter should be between $^{29}/_{64}$ and $^{17}/_{32}$ inch, or the equivalent area; the i. m. e. p., the fuel economy, and combustion knock at 1,500 r. p. m. favoring the smaller passage size.

(*b*) The size of the connecting passage becomes less critical as the engine speed is decreased. It is therefore possible to select a passage size for maximum operating speed and still have good performance at the lower speeds.

(*c*) The friction mean effective pressure due largely to throttling losses was excessive when a passage diameter of less than $^{29}/_{64}$ inch was employed; however, for a passage diameter equal to $^{29}/_{64}$ inch, the friction mean effective pressure was acceptable and the rate of decrease with increase in passage area became much less.

(*d*) It was impossible to obtain both high performance and combustion-pressure control with any combination of variables tried in this investigation.

The general conclusions are:

From the results of this investigation of the prechamber type of cylinder head, several optimum conditions are evident. For maximum performance of this engine, which has a 5-inch bore and a 7-inch stroke, the compression ratio should be between 15.5 and 17.5, the prechamber should be relatively as large as possible, disk-shaped, and connected to the cylinder clearance by a single passage the area of which is determined by the highest engine speed. Entering the chamber tangentially, the passage should cause a strong rotational air flow and, upon entering the cylinder, should be flared to spread the issuing gases over the piston crown. The injection system should deliver full-load fuel with atmospheric induction in the shape of a narrow conical spray of high penetration requiring approximately 20 crankshaft degrees for injection. This spray should be directed across the disk chamber, with the air flow, toward the mouth of the connecting passage. The spray direction greatly affects the engine performance. Considerable improvement in engine performance and combustion knock can be obtained by boosting.

As all of the clearance should be in the prechamber, proper valve timing and scavenging are prohibited, a condition which limits the specific output.

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS,
LANGLEY FIELD, VA., *July 7, 1936.*

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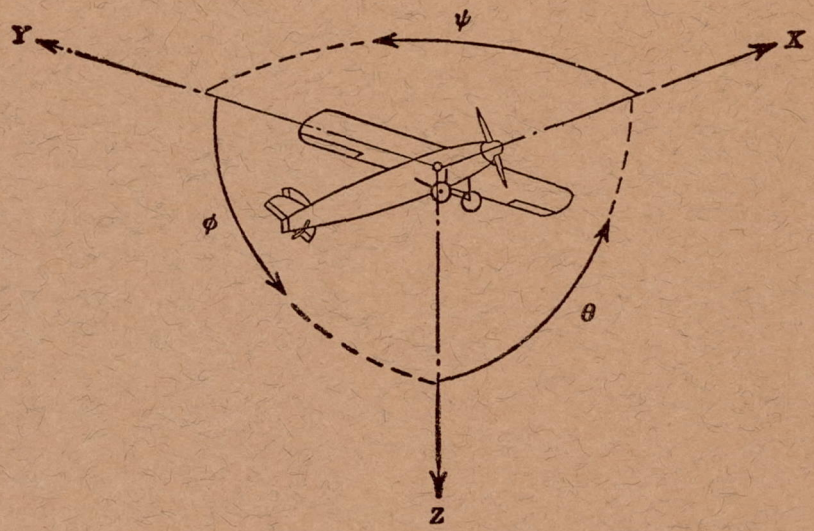

Positive directions of axes and angles (forces and moments) are shown by arrows

Axis		Force (parallel to axis) symbol	Moment about axis			Angle		Velocities	
Designation	Symbol		Designation	Symbol	Positive direction	Designation	Symbol	Linear (component along axis)	Angular
Longitudinal	X	X	Rolling	L	$Y \longrightarrow Z$	Roll	ϕ	u	p
Lateral	Y	Y	Pitching	M	$Z \longrightarrow X$	Pitch	θ	v	q
Normal	Z	Z	Yawing	N	$X \longrightarrow Y$	Yaw	ψ	w	r

Absolute coefficients of moment

$C_l = \frac{L}{qbS}$ (rolling) $\quad C_m = \frac{M}{qcS}$ (pitching) $\quad C_n = \frac{N}{qbS}$ (yawing)

Angle of set of control surface (relative to neutral position), δ. (Indicate surface by proper subscript.)

4. PROPELLER SYMBOLS

D, Diameter

p, Geometric pitch

p/D, Pitch ratio

V', Inflow velocity

V_s, Slipstream velocity

T, Thrust, absolute coefficient $C_T = \frac{T}{\rho n^2 D^4}$

Q, Torque, absolute coefficient $C_Q = \frac{Q}{\rho n^2 D^5}$

P, Power, absolute coefficient $C_P = \frac{P}{\rho n^3 D^5}$

C_s, Speed-power coefficient $= \sqrt[5]{\frac{\rho V^5}{P n^2}}$

η, Efficiency

n, Revolutions per second, r.p.s.

Φ, Effective helix angle $= \tan^{-1}\left(\frac{V}{2\pi r n}\right)$

5. NUMERICAL RELATIONS

1 hp. = 76.04 kg-m/s = 550 ft-lb./sec.
1 metric horsepower = 1.0132 hp.
1 m.p.h. = 0.4470 m.p.s.
1 m.p.s. = 2.2369 m.p.h.

1 lb. = 0.4536 kg.
1 kg = 2.2046 lb.
1 mi. = 1,609.35 m = 5,280 ft.
1 m = 3.2808 ft.